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INVESTIGATION OF ELECTROHYDRAULIC PULSE MOTORS FOR AIRCRAFT UTILITY FUNCTIONS

LOCKHEED-GEORGIA COMPANY
A DIVISION OF LOCKHEED AIRCRAFT CORPORATION
MARIETTA, GEORGIA 30063

MAY 1977

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AIR FORCE AERO-PROPULSION LABORATORY
AIR FORCE WRIGHT AERONAUTICAL LABORATORIES
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WRIGHT-PATTERSON AIR FORCE BASE, OHIO 45433

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Kenneth E. Binns

KENNETH E. BINNS, Project Engineer
Fluid and Mechanical Systems
Vehicle Power Branch

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STEPHEN P. CONDON, Major, USAF
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The purpose of this electrohydraulic pulse motor (EHPM) technology-development project was to investigate feasibility and applicability and demonstrate capability of the EHPM concept for mating the power of hydraulics with the "intelligence" of the digital computer for use in future aircraft utility systems. Recent dramatic developments in electronic circuit manufacturing necessitates the review of control concepts for the purpose of redefining the optimum mix of technologies. The significant reduction in cost, size, and weight and increase in reliability of digital computers offers the potential for performing the bulk of

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control functions within the electronic circuit in lieu of the hydraulic circuit. The EHPM concept is highly compatible with this concept, but existing units are used in machine tools and are not configured to achieve maximum power and minimum weight as needed for aircraft use.

In this program the designs of EHPM's for aircraft use was investigated; an EHPM was designed and constructed using an aircraft type hydraulic motor; an input system consisting of a micro computer, an interface board, input and output position encoders, an electric pulse motor (EPM) driver, and a software program was designed and constricted; the EHPM and the input system were tested as components and as a system; the system was used to control the C-5 iron bird flap system with simulated loading; the EHPM unit was subjected to a durability test; and a survey of various aircraft subsystems where the EHPM could be a viable alternative was accomplished.

Testing revealed several problems with the current design, but all problems were considered to be correctable with state-of-the-art techniques.

The concept is considered to be a viable alternative approach to utility and secondary flight control systems, and the principal payoff will be improved reliability and maintenance cost.

PREFACE

The work reported here was performed by the Lockheed-Georgia Company, A Division of Lockheed Aircraft Corporation, Marietta, Georgia, under Air Force Contract F33615-75-C-2005, "Investigation of Electro-Hydraulic Pulse Motors for Aircraft Utility Functions." The Air Force Project Manager was Mr. Kenneth E. Binns and the Lockheed Project Manager was Mr. Edwin W. Rumrill.

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Mr. Frank D. Lewis, Technical Assistance

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SUMMARY

The electrohydraulic pulse motor (EHPM) presently being used for machine tool control offers an effective interface between the "intelligence" of the low power digital computer and the "strength" of high power hydraulics. The machine tool systems, however, are not suitable for aircraft because of weight, cost, and volume. Advances in electronic circuit technology are leading to values which will make the input concept a viable appraoch for aircraft utility and secondary flight control systems. The EHPM device needs adapting to aircraft use to achieve the horsepower to weight values for aircraft hydraulic drives, and the purpose of this program was to develop an aircraft EHPM.

Existing EHPM technology was investigated including the state-of-the-art of the electrical pulse motor (EPM) - a key element of the concept. Potential uses in aircraft utility and secondary flight control systems were studied. A prototype EHPM and a breadboard input system using an Altair 8800 microcomputer were designed, constructed, and tested.

Testing consisted of bench tests, iron bird testing wherein the system was used to drive the C-5 flap simulator with simulated loading, and durability testing.

Several problems were encountered with the prototype unit; the EPM torque is insufficient for the design, and full speed was not achieved in one direction; the design is sensitive to return pressure but can be altered to reduce the sensitivity; failure of the plating on the spool apparently caused a seizure of the spool; sleeve retainment was not adequate; and non-jamming thread stops need to be used on the drive threads.

All of the problems encountered are considered to be correctable by redesign. Additionally it was concluded that for aircraft purposes a pilot concept wherein the EPM drives a very small pilot valve which hydraulically positions the main valve can be more appropriate than the direct drive. This approach allows use of a smaller EPM and EPM drive circuit. Reduction of speed between the hydraulic motor and the valve will allow the EPM to operate at lower speeds on a higher level of its torque curve. Resolution reduction is no obstacle as existing EHPM resolution exceeds aircraft requirements by several orders of magnitude.

The EHPM concept is considered to be a viable alternative approach to utility and secondary flight control systems especially where multiple actuators require synchronizing. The principal payoff is expected to occur in improved reliability and maintenance costs due to a significant reduction in the quantity of moving parts.

SECTION I
INTRODUCTION

Toward the goal of providing lighter, less costly, and more reliable aircraft actuation systems, electro-hydraulic pulse motors (EHPM) offer the promise of a logical interface between the airborne digital computer and the loads carried by large hydraulic power actuation systems. The practical application of an EHPM system is made possible by recent advances in electrical circuitry and control technology.

In Japan and Europe, the "stepper" was adapted to openloop control of machine tools, after extensive use of electrical stepping motors in data processing and instrumentation equipment. When it appeared that needed acceleration rates could not be achieved, the electrical stepper was mated with the hydraulic motor to increase driving torque, and the EHPM was created. Preliminary analysis indicates a promising future for the application of EPHM to aircraft systems. It is a highly attractive concept for mating the power of hydraulics with light weight low cost electronic control circuits and the "intelligence" of the digital computer. Although the initial EHPM's have been used successfully in the machine-tool industry to control motion in exacting machine operations, the existing hardware is not suitable for aircraft use. Constraints of weight, volume, duty cycle, and environmemt will not allow aircraft use of machine-tool industry developed units.

The current program is aimed at evaluating and adapting the EHPM for control and power of aircraft utility systems. The program performs a preliminary analysis of available equipment and the interfaces of the various components of EHPM actuation system including the input control system, the pulse motor, control valve and hydraulic motor. A prototype EHPM was built and tested to drive a flap actuation system controlled by a digital input control system.

SECTION II
ANALYTICAL STUDY

State-of-the-Art and Available Hardware Survey

The EHPM - The Reference 1 magazine article entitled, "Guide-to-Performance and Specifications of EHSM's", is a treatment of the state-of-the-art. It provides a chart comparing the performance of units which are produced by six manufacturers. Fujitsu units are nearest to aircraft requirements with models utilizing 2,000 psi and involving speeds to 4,000 RPM. Sizes available from Fujitsu are in the power range of 0.8 to 20 horsepower. Since aircraft hydraulic motors operate at higher speeds and pressures to achieve minimum weight, the optimum arrangement, as will be noted later in the design section, must involve a gear reduction between the electrical stepping motor (EPM) and the hydraulic motor.

The Electrical Stepper Motor (EPM) - The electrical pulse motor is a critical component of the EHPM package. The hydraulic valve and the hydraulic motor present no critical design problems except as pertains to reducing the torque and speed required of the electrical pulse motor. A key effort in developing the EHPM to aircraft use is to obtain an electrical pulse motor or an arrangement that will permit use of the hydraulic motor up to its maximum capability. As a first step in this task, a survey of available steppers was conducted. The Manufacturer's Chart, Figure 1, represents the result of this survey. Parameters chosen for inclusion in the chart are intended to convey a "quick look" at stepper capability.

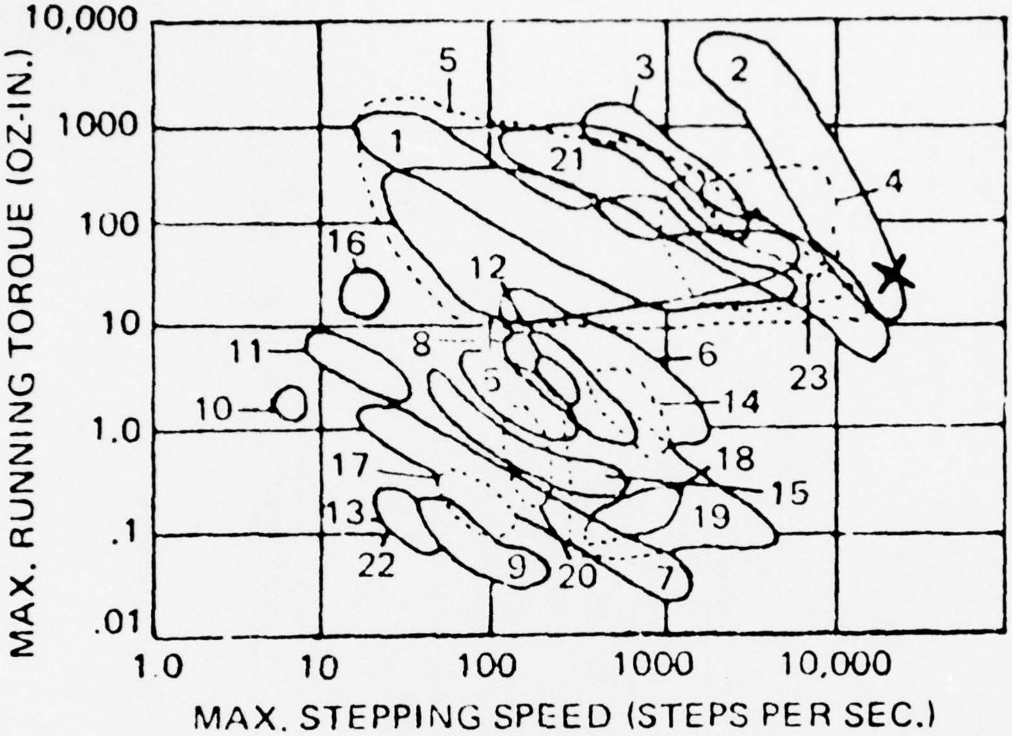
The "Baloney" Chart, Figure 2, reproduced from Reference 2 article, summarizes the state-of-the-art for stepper motors in regard to the speed versus torque characteristics. Selection of hardware, requires the use of in-depth performance data which are appropriately provided on curves provided by the supplier.

Figure 3 illustrates a manufacturer's presentation of specification for a line of stepper motors. The series presented is the Fujitsu HI-PM line and the HI-PM 0 is the selected unit for the test program. While data were gathered on many other manufacturers and on various hybrid types such as "flexspline", "synstep", and "nutating gear", the charts represent data on only those units which appear to offer performance close to that required for driving an EHPM; i.e., the top performers for torque and RPM which are believed to represent the best state-of-the-art. The two types of units in the chart are Variable Reluctance (VR) and Permanent Magnet (PM) types. These two types offer appropriate characteristics for use in aircraft EHPM's, including mechanical simplicity, high RPM, and low weight.

An important difference in stepper performance needs for aircraft applications and conventional applications lies in the area of resolution and shaft speed. Most uses require a small step but a high step rate. In the machine tool applications, a small increment of control is necessary, and one step is used to cause a motion of .0001 inch. A step rate of 20,000 pulses/second offers a tool translation maximum rate of

Manufacturer	Type	Motor Frame Sizes	# of Mod	Steps per Second at Zero Load Torque: Maximum Slewing Rate	Steps per Second at Zero Load Torque: Pulse to-Step Fidelity	Step Size (deg.)	Accuracy (deg.)	Peak Holding Torque (oz.in.)	Rotor Inertia (gcm^2)	Voltage Rating
Computer Devices (Rapid/Syn)	VR	8-23	18	500-15,000	400-2,000	2.25-15	0.06-0.45	1.0-50	0.18-350	6-28
	PM	8-34	40	180-2,000	175-1,100	1.8-90	0.05-2.7	0.8-500	0.3-1,850	2-35
Eastern Air Devices	VR	20	6	2,000-6,000	450-650	-15	-	35-65	10-20	-28
	PM	23-34	10	500-20,000	300-950	-1.8	-	70-265	120-1,100	2-35
IMC Magnetc (Tormax)	VR	8-25	24	375-6,000	350-1,050	7.5-15	±0.2-±0.4	0.35-110	0.1-15	5.5-28
	PM	5-34	24	150-30,000	105-900	1.8-90	±0.05-±1.0	0.25-60	0.19-400	5.5-48
Novatronics (Muirhead)	VR	8-23	8	680-2,560	80-1,012	1.8-15	0.05-0.75	0.9-22	0.18-3.8	6-28
	PM	8-23	22	90-1,200	70-720	1.8-90	0.05-4.5	0.65-53	0.7-187	6-28
Sigma	PM	9-20	90	160-10,000	-2,000	0.72-22.5	±0.05-±0.90	1.0-1,000	0.6-11,000	2-48
Singer Kearfott	VR	8-40	24	-	120-2,000	1.8-90	0.05-2.5	0.5-175	0.28-110	28
	PM	8-40	11	-	100-390	1.8-90	0.05-2.5	0.3-22	0.26-23	28
Superior (Slo-Syn)	PM		100	-20,000	-5,000	0.72-1.5	±0.05-±0.09	53-2,100	0.03-2.75	1.3-28
USM ICON (Fujitsu)	VR		6	-16,000	-2,600	1.2-4.5	-0.67	30-480	34-1,356	Varies with drive circuitry
Warner	VR	20-60	50	3,000-16,000	300-1,300	1.8-15	±0.09-±0.33	3[illegible]-1,100	8.47-5,380	1.2-28

Figure 1 Available Stepper Motor Characteristics



Key:

(19) American Electronics, Inc.
(14) Automation Development Corp.
(9) Bowmar Instrument
(20) Cedar Engineering Co.
(24) Computer Devices
(7) Clifton Precision Products
(21) Dahmen-Burnett Electronics
(8) General Precision (Kearfott)
(16) Giannini
(22) Haydon Switch & Instrument
(10) Heinemann Electronics
(2) Icon Corp.
(15) IMC Magnetics
(11) Ledex, Inc.
(23) MesurMatics Electronics
(12) Muirhead Instruments, Ltd.
(13) North American Philips Controls
(5) Sigma Instruments, Inc.
(1) Superior Electric Co.
(3) USM Corp.
(18) Vernitron Corp.
(4) Warner Electric Brake & Clutch Co.
(17) Weston Transicoil Div.
(6) Wright Machinery Co.

Figure 2 Performance Comparison of Available Stepper Motors from Electromechanical Design, Reference 2

Kinds / Item \ Motor Type	Hi·PMs (High Pulse Motors)									
	Hi·PM 0	Hi·PM 5		Hi·PM 10		Hi·PM 20		Hi·PM 30		Hi·PM D
Angular Increment per Pulse deg.	1.2	1.5		1.5		1.2		1.2		0.36
Maximum Running Pulse Rate pps	16,000	16,000		16,000		16,000		16,000		16,000
Output Power (Approximate) HP		class A	class B	class A	class B	class A	class B	class A	class B	
	0.13	1.5	0.8	2.5	1	3	1.5	3.5	2	3
Output Torque kg.cm (oz·in)	2.5 (35)	30 (420)	30 (420)	70 (970)	70 (970)	100 (1,400)	100 (1,400)	150 (2,100)	150 (2,100)	330 (4,600)
	at 100pps									
	4 (55)	27 (370)	22 (310)	60 (830)	45 (630)	82 (1,150)	67 (930)	115 (1,600)	100 (1,400)	275 (3,800)
	at 8,000pps									
	3 (42)	25 (350)	15 (210)	50 (700)	20 (280)	65 (900)	35 (490)	80 (1,100)	50 (690)	220 (3,050)
	at 16,000pps									
Allowable Load Inertia $kg \cdot cm \cdot sec^2$ ($lb \cdot in \cdot sec^2$)	4×10^{-4} (3.5×10^{-4})	1×10^{-2} (0.86x1[illegible])		[illegible]10^{-2} ([illegible]$\times 10^{-2}$)		4×10^{-2} (3.5×10^{-2})		8×10^{-2} (6.9×10^{-2})		4×10^{-1} (3.5×10^{-1})
Rotation Accuracy (step) (Maximum Cumulative Errors)	±0.2	±0.2		±0.2		±0.2		±0.2		±0.2
Approximate Weight kg (lb)	3 6.6	14 30.8		28 61.6		40 88		60 132		60 132

Figure 3 Specifications for ICON (Fujitsu) Hi - PM Series

2 inches/second or 120 inches/minute. In aircraft applications, it is reasonable that one pulse can represent a much larger displacement, such as three orders of magnitude greater, but the speed requirements for hydraulic motors are also higher. A 300 pulse/rev stepper, stepping at 20,000 pulses/second provides 4,020 RPM. This RPM capacity, important to aircraft applications, is not reflected directly in the Figure 1 Characteristic Chart but may be calculated as follows:

$$\text{RPM}\left(\frac{\text{revs}}{\text{min}}\right) = 60\,\frac{\text{PPS (Pulses/Second)}}{\text{PPR (Pulses/Revolution)}}$$

Stepping motor technology has initiated a new language which must be introduced in order to apply the technology. Some selected terminology is provided in the section on stepper motors and in Appendix A.

Input Control System - As previously stated, the advancement in the state-of-the-art of electronic circuit manufacturing is a prime factor that makes this program applicable. Complex control functions can be handled with standard circuit modules at low cost and weight and with good reliability offering the potential for removing control functions from the hydraulic system where the hardware is heavy, specialized, and involves complex mechanical moving parts subject to interactions, seizing, leakage, etc. As an example, it often requires from 7 to 10 hydraulic valves to control a utility actuation system in "bang-bang" fashion whereas the EHPM concept offers control with one valve, and performance is optimized by control of acceleration/deceleration and speed at any position of the load. The capability to free the system of pressure surges and to alter performance characteristics by means of software changes in the computer in lieu of valve hardware changes are attractive considerations. The fact that additional input parameters can be read into the control functions usually at low cost per parameter and at various stages of system development is an important plus for the concept. Although this program is directed toward the development of the EHPM, it is technology advancement in the input control that makes it feasible.

The input control system consists of two basic control units 1) The microcomputer and 2) The stepper motor drive circuits. Microcomputers are available in many sizes and capabilities and standard microcomputers may be used for development of EHPM input controls. The stepper motor drive circuits are designed and developed for the particular design of the stepper motor and are available from the stepper motor manufacturer. Development of EHPM input control systems can utilize these standard microcomputers or stepper motor drive circuits, however final application of the control system may package the complete input control system in a single unit. A more complete discussion of these important control units follows in component analyses and in Appendix A.

Component Analysis

Input Drive Systems - An EHPM is basically a device that transforms a stream of low power electrical pulses into high power mechanical motion such that the total travel is proportional to the quantity of pulses, the rate is proportional to the frequency of the pulses, and acceleration is proportional to change in frequency of the pulses. The input control system must, therefore, read a signal from the crew, or some automatic

control which calls for a position and rate of an actuated device. It can also "read in" other control parameters such as status of other systems or aircraft performance data. It must interpret these signals, transform them into a stream of pulses which are acceptable to the EHPM, and deliver them in the correct rate and quantity. The input control signals need to be at low power levels except where high power is required into the stepping motor.

The block diagram of Figure 4 illustrates the elements of a simple input control system. Elements A, B, and C are low power circuits in order to utilize advance technology Large Scale Integration (LSI) circuit modules. These elements transform the input commands into the required pulse frequency and quantity and select the proper coil(s) of the EPM to energize. Element D amplifies the signal to the coil(s) being energized to the power level required by the EPM. Element E of Figure 4 can be used to sense when the regime for successful open loop control is about to be exited. A new regime can be entered and the system operated at reduced speed. Element E can also be used to operate the system strictly closed loop.

Elements A and B (containing the microcomputer) control the input signal into the sequential logic element which is a stream of pulses upon which the EPM responds on a one-to-one basis. Control of the pulse stream to achieve direction, displacement, ramping, reaction to any feedback, or any control function will have been performed before the stream reaches the sequential logic element. It is, therefore, the function of the "front" elements, which are defined as those between the crew and the sequential logic element, to control the pulse timing, i.e., frequency and change in frequency; the quantity of the pulses; and to select the line on which to put the pulses to apply forward or reverse rotation. The elements of the input system up to the sequential logic element can be of various configurations and the appropriate approach is a strong function of the system application; i.e., input and output complexity; number of channels; flexibility requirements; and projected growth or uncertainty for change. For the purpose of this study, however, the most flexible and the highest state-of-the-art which utilizes the standard LSI module approach -- digital computer control -- is of prime interest and the program is geared to this approach. There are several microcomputers on the market which can perform the required function of this portion of the input system. The packaging of these off-the-shelf units is not representative of ultimate aircraft packaging, however, the circuit elements are representative and on a functional basis, this hardware is appropriate for the program. It is noted that final packaging of the input system was specifically excepted as an objective for the program.

The "final" elements of the input system, which are defined as the sequence logic and the pulse driver, respond to the input train of pulses as they are received on either the forward or the reverse line so as to energize the stepper motor windings with high power in the proper sequence to cause the motor to step in the chosen direction. Each low power input pulse causes one or more motor coil(s) to be energized with high power.

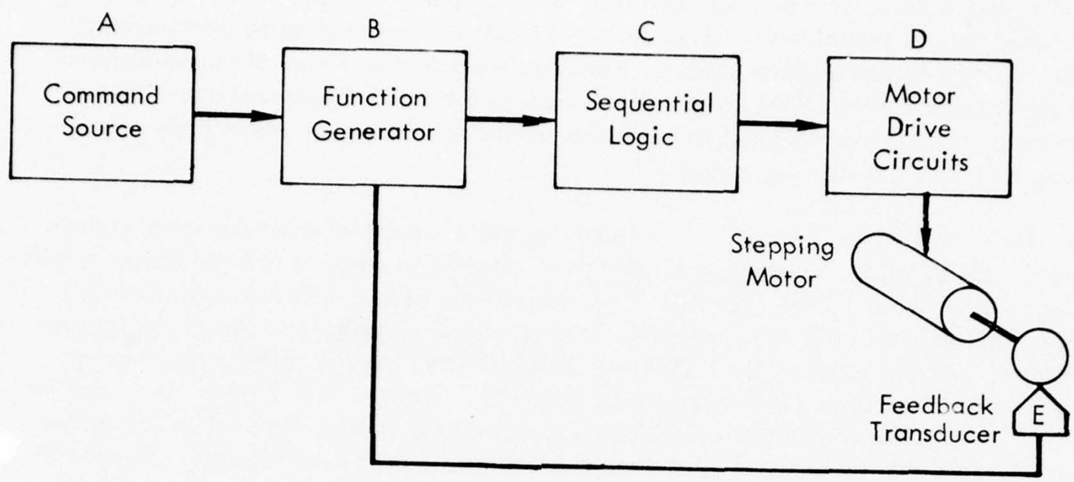


Figure 4 Simplified Stepping Motor Control System

The final elements provide low power circuitry to perform the sequencing -- the selection of which coils to energize -- and a solid state relay for each coil of the motor to switch in the high power.

Off-the-shelf hardware is available from most of the stepping motor suppliers to perform the function of the final elements. This hardware can be procured either in a chassis for rack mounting or on circuit cards. As in the case of the forward elements, the prototype packaging for the final elements will not be representative of an appropriate aircraft configuration.

The microcomputer concept is an important development in respect to future aircraft control systems using the EHPM. It is, therefore, appropriate to define the term and reflect on its relationship to the EHPM.

The basic arrangement of a digital computer is shown in Figure 5. It is noted that three basic elements are provided: A CPU, a ROM, and a RAM. The Central Processor Unit (CPU) controls the transfer of information and performs calculations, the Read Only Memory (ROM) is fixed memory and the Random Access Memory (RAM) is variable memory which will be altered continuously during system operation. These functions are the basic building blocks of a control system. Each can be thought of as separate circuits on a chip. They can be standard off-the-shelf hardware. A given microcomputer may consist of a number of these standard modules assembled into a black box. The number of ROM's and RAM's required depends upon the complexity of the control task. It should be noted that the ROM contains the fixed program that adapts the standard hardware to the special control task. For example, a black box to control the flap system may be exactly like a black box to control the forward cargo door complex. The only difference is the way the ROM modules are programmed. ROM can be reprogrammed by removing the module from the aircraft, erasing the existing program, and energizing a new program into the chip.

The term microcomputer was coined merely to connotate size, both physically as well as functionally; i.e., size of memory and program capability. The concept is extremely important to the EHPM concept and to future aircraft controls because of low cost, small size, and high reliability. As one author aptly stated: "The day of the matchbox computer draws near". That is, small highly flexible digital controls will be used for the simpler control tasks.

Predictions of reduced cost and size, and higher reliability of computer modules have to a large degree already "come to pass" and the evidence sits on many engineer's desk -- the "electronic slide rule". The elements of these devices are the types of circuits that will provide the control functions for many EHPM input systems.

Electrical Pulse Motor (EPM) - A stepper motor is an electromagnetic incremental actuator which converts electrical pulse inputs to output motion. Two common acronyms are ESM (Electrical Stepper Motor) and EPM. EPM is used in this report.

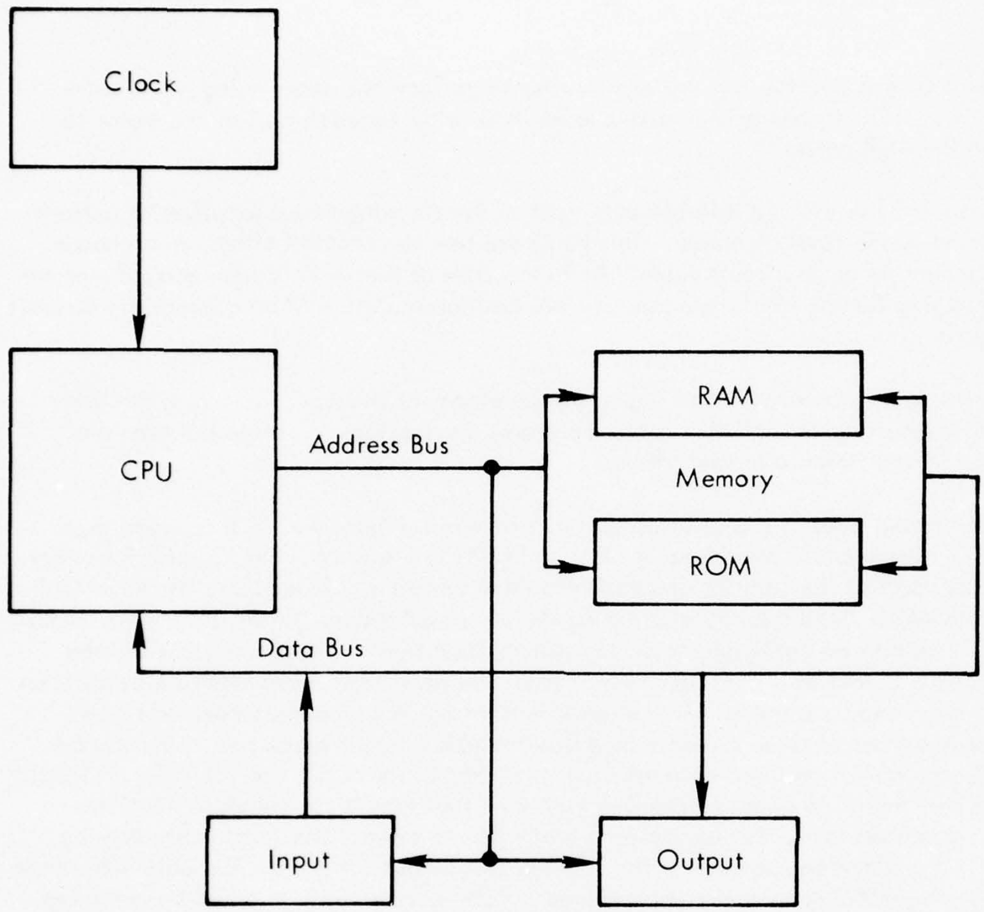


Figure 5 Organization of Microcomputer

When energized electrically in a programmed manner, it indexes incrementally. When operated within its capability the output steps are always equal in number to the number of input pulses. Each pulse advances the rotor shaft and latches it magnetically at the positions to which it is stepped. The motor provides rapid acceleration, stopping, and reversal.

An electronic circuit is required to transform the input pulse train into this sequential energizing pattern. The power level of the control pulse train is very low compared with the power level of the EPM coils so this circuit also involves amplification. This circuit is commonly referred to as the driver.

When operating in the slew mode, that is, running at a speed higher than its instantaneous start-stop or reversing speed range, the stepper motor will maintain synchronism with the pulse train. To start, stop, or reverse in this range of operation, the motor must be accelerated and decelerated to and from the slew speed. Thus, for acceleration, the frequency of the pulse train should be increased from the starting speed up to the final frequency, and for deceleration, the frequency of the pulse train must be decreased.

A list of stepper motor characteristics which are important to the design and application of an EHPM are as follows:

- No accumulative error.
- Predictable and consistent performance when within limits of capability.
- Controls are digital; easily adapted to computer control; respond to pulse commands.
- Simple; only two bearings in mechanical contact; no maintenance.
- Usable open loop with desirable features of a feedback system; relatively clean null, no drift.
- Quiet
- Free from contaminants, not sensitive to contamination.
- Can be repeatably stalled without damage.
- Bidirectional rotation.
- Fixed step angle or increment of motion.
- Low efficiency.
- Limited ability to handle large inertia loads.
- Friction loads increase position error, error is non-accumulative.

Stepper motor technology involves its own unique terminology. Some of the more significant terms related to torque are discussed in this section and others are discussed in the Glossary of Terms, Appendix A.

Holding (Static) Torque - The holding torque curve is a fundamental torque characteristic of a stepper motor. The origin of the curve corresponds to a motor energized and at rest at any of its step positions. This curve shows the holding torque versus rotor angular displacement from the step position. This torque acts in a direction to force the rotor back to and hold it in the zero-torque step position.

The holding curve is one segment of the total torque-function curve corresponding to each phase of the motor. All other segments of these phase-torque-function curves are formed from this one holding torque curve, or its images formed by rotation about the vertical and horizontal axes. Thus, the holding torque curve is all that is necessary to completely determine the instantaneous torque of the motor under all possible static conditions of excitation and rotor position. All other torque characteristics, static or dynamic, have their origins in this holding torque curve.

Five segments of the holding torque curve for the HiPMO are illustrated in Figure 6 The displacement data are accurate for the HiPMO null positions, but the shape of the curve is arbitrary to illustrate the general appearance of a holding torque curve. Data to apply torque versus displacement numbers to this curve were not available in supplier literature nor was testing done to obtain it. If a system is to be modeled for computer analysis, the characteristic illustrated by this curve would need to be measured by tests wherein the motor is displaced a measured amount and the restoring torque measured for a set of displacements, encompassing $\pm 6^\circ$ for the HiPMO.

Pull-out Torque - The pull-out torque (torque-speed) curves, Figure 7, indicate the maximum steady-state friction torque which can be applied as a load on the motor at the corresponding speeds, or stepping rates, without pulling the rotor out of synchronism with the input pulse train and stalling the motor.

It is important to understand that the pull-out torque curves have no counterpart in the conventional motor field. They do not define operating points, nor are they representative of a transfer relationship. They simply define the region of torque-speed combinations inside which the motor will operate satisfactorily and outside which it will not operate at all, for a given set of excitation and control conditions.

A limitation to the significance of these torque-speed curves is that they assume constant velocity at a given speed. This is only true at stepping rates of several hundred steps per second, depending upon the motor, inertia load and control. A stepper motor is, in fact, starting and stopping at low step rates and changing instantaneously the step rate to a slightly higher step rate. Then, in this "stepping-mode" range, the motor must exert accelerating and decelerating torque on its own internal inertia and the coupled inertia in addition to the continuous torque implied by the speed-torque curves.

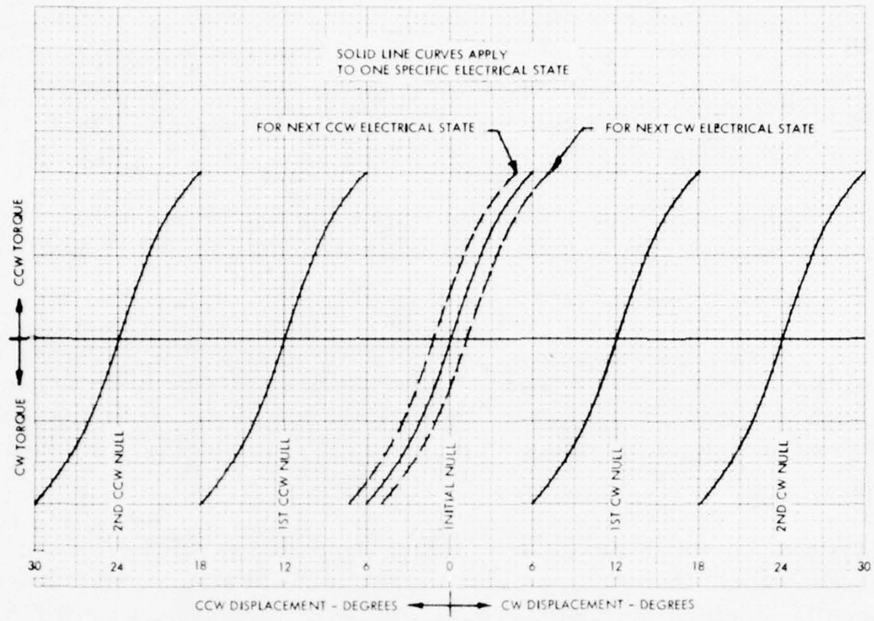


Figure 6 Family of Holding Torque Curves

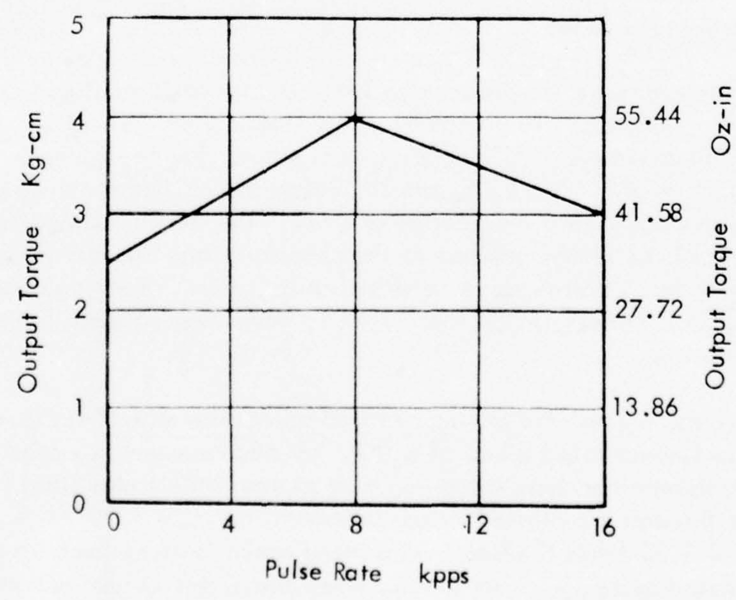


Figure 7 Pullout Torque Vs. Pulse Rate for Fujitsu Hi - PMO

In the open-loop mode, the motor must be operated below its maximum dynamic torque capability to ensure that the motor does not stall or miss steps. In the closed-loop (feedback or self-commutating) mode, the motor can be operated at, or near, its maximum dynamic torque. Motor heating under dynamic conditions is normally not a consideration. Motor ratings are determined by the power the motor can dissipate under static conditions with normal phases energized. When a motor is running, the current varies little with or without a load.

Stepper Motor Performance Presentation - In order to further illustrate performance characteristics of stepper motors, ICON's performance curves for the HiPMO are shown in Figure 8.

In Figure 8 (Sheet 1), the ordinate is the minimum acceleration/deceleration time constant and the abscissa is load inertia. The acceleration/deceleration time constant or time referred to means the value of T_a and T_d when pulse rate f of the input pulse train, as shown in Figure 8 (Sheet 2) rises and falls exponentially or linearly, respectively.

Hydraulic Motor - Aircraft hydraulic motors have been used to drive utility actuation systems for many years. With inherent characteristics of low weight, high power output and versatile operation for continuous, intermittent, reversible or stalled duty cycles, it enjoys a wide range of applications. Aircraft hydraulic motors are designed to specification MIL-M-7997B unless requirements are modified by a detail specification. The majority of developed motors are designed to operate in a 3,000 psi pressure system with oil temperatures from -65°F to 275°F. For military applications, MIL-H-5606 fluid is used. It is intended that the standard aircraft hydraulic motor with years of experience and development background will be used for the design of the integrated electro-hydraulic pulse motor (EHPM).

Standard aircraft hydraulic motors are available in two basic configurations: bent axis and in-line. The bent axis design has been the standard of the industry for many years. The newer in-line design is receiving wide acceptance due to its simple compact design and low cost and weight. In-line hydraulic motors are an outgrowth of proven pump design and are available in a wide range of sizes. The in-line design has been selected for development of the EHPM because of the simplicity and compactness of the complete assembly pulse motor, control valve, and hydraulic motor. Both major suppliers of hydraulic pumps and motors, ABEX and Vickers, have sizes ranging from .02 to 8 cubic inches per revolution.

Available sizes and normal and maximum rated speeds are shown in Figure 9. A theoretical, continuous rated speed of $6{,}000/\sqrt[3]{\text{displacement}}$ is added as a reference. This rating was determined from a log-log plot of available motor sizes and speeds by drawing a line through the plotted values as shown in Figure 10. A maximum intermittent speed of 1.25 times the continuous rated speed is also shown on this figure. It can be noted from Figure 10 that motor sizes above two cubic inch per revolution have higher continuous ratings and approach values of approximately $8{,}000/\sqrt[3]{\text{displacement}}$. This fact is indicative of the development of high speed, large displacement pumps and motors for late model aircraft.

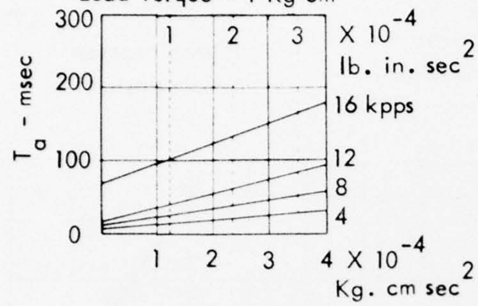
Acceleration

Linear acceleration

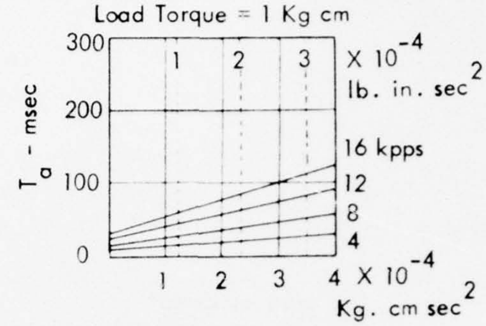
Exponential acceleration

Refer to Sheet 2 for Definition of Ta and Td

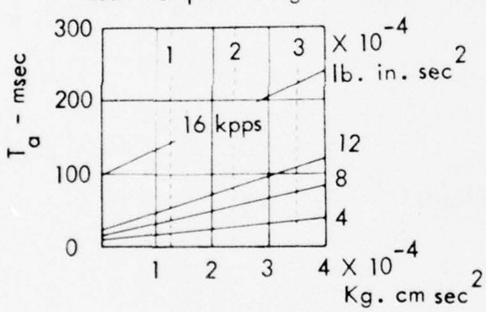
Load Torque = 1 Kg cm



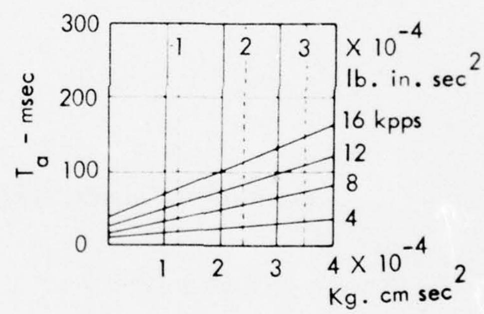
Load Torque = 1 Kg cm



Load Torque = 2 Kg. cm



Load Torque = 2 Kg. cm

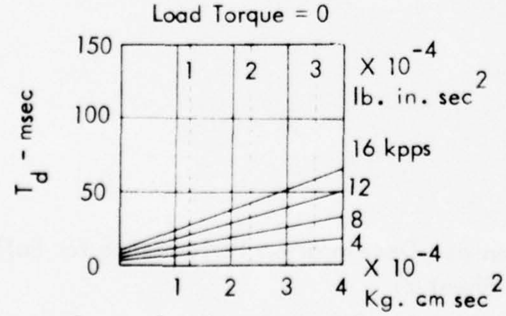


Deceleration

Linear deceleration

Exponential deceleration

Load Torque = 0



Load Torque = 0

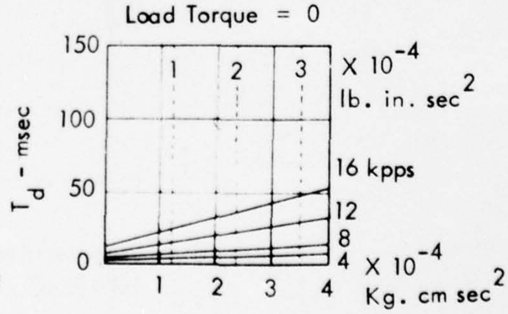
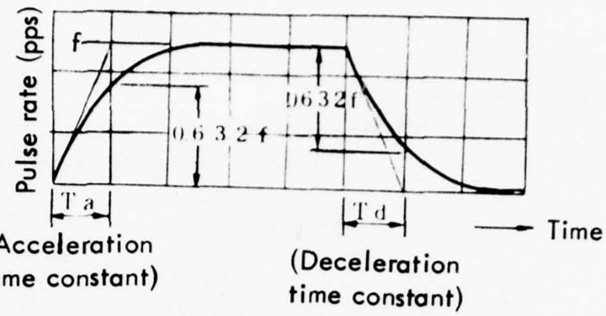
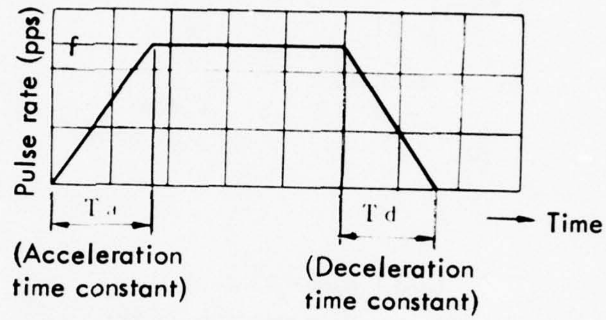


Figure 8 Acceleration and Deceleration Performance for Fujitsu Hi - PM 0 Sheet 1

In case of exponential acceleration/deceleration



In case of linear acceleration/deceleration



Exponential Acceleration Equation:

$$R = f\left(1 - e^{\frac{-t}{Ta}}\right)$$

Exponential Deceleration Equation:

$$R = f\,e^{\frac{-t}{Td}}$$

Figure 8 Acceleration and Deceleration Performance for Fujitsu Hi-PMO (Sheet 2)

Displacement In^3/Rev			Rated Speeds	
Abex	Vickers	Normal	Maximum	$6000/\sqrt[3]{DISP}$
	.020	18000	22500	22104
	.030	18000	22500	19309
	.060	15000	18750	15326
0.10		10000	12000	12927
	0.11	12500	15600	12522
0.18		8500	11000	10627
	0.22	10000	12500	9939
0.31		8000	10000	8865
0.40		6500	7500	8143
	0.44	8000	10000	7889
0.67		6200	7200	6857
	0.75	7000	8750	6604
1.00		6000	7000	6000
	1.15	6000	7500	5727
1.37		5100	6000	5402
	1.50	6000	7500	5241
1.77		3750	4500	4960
2.00		5800	7200	4762
	2.05	5650	7100	4723
2.40	2.40	5300/4000	6600/4800	4481
3.00	3.00	5000/5400	6250/6600	4160
4.40		5000	6000	3662
6.5		4250	5300	3215
	7.14	3700	4600	3116
8.5		3750	4500	2940

Figure 9 Hydraulic Motors, Displacement and Speed

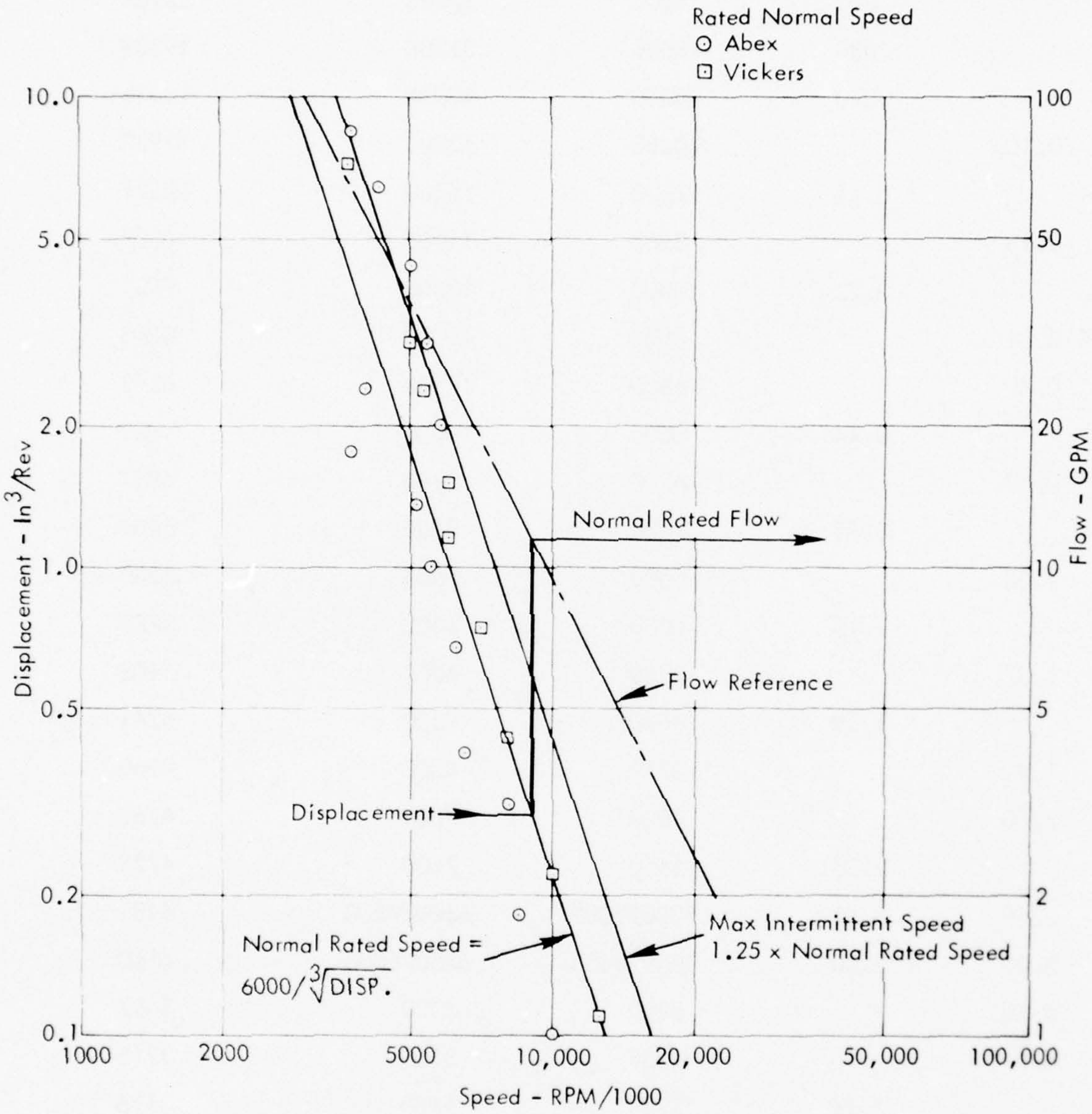


Figure 10 Hydraulic Motor Speed & Flow Characteristics

Standard mounting flanges on hydraulic motors interface with actuation system gearboxes. These standards are listed in the motor specification MIL-M-7997B. The following table lists the military standards with torque, weight, and overhung moment limitations. Detail dimension and requirements can be found on the respective standards.

			Accessory Maximum	
Motor Mount	Gearbox Mount	Torque in.-lb	Weight lb	Moment in.-lb
AND10260	AND20000	100	6	25
AND10261	AND20001	500	30	150
AND10262	AND20002	2,500	65	400

At 3,000 psi, the torque limitations permit motor sizes of maximum displacement of 0.2 in.^{3}/rev on AND10260, 1 in.^{3}/rev on AND10261, and 5 in.^{3}/rev on AND10262.

In the EHPM, the control valve housing will replace the normal motor valve plate and port cap. The dimensions of the valve plate are peculiar to the particular motor design and are critical to proper pressure balances. Therefore, these dimensions will not be detailed in this report. The dimensions should be identical to the present motor port cap in the area of the motor valve plate. The valve plate must be a hardened steel material and the valve housing aluminum (for minimum weight). Typical valving surfaces of hydraulic motors are illustrated in Figure 11. Designs utilizing a thin intermediate valve plate and aluminum port caps are common where port caps are more complex and weight is critical. Attachment of the valve housing to the motor housing can be made by the same bolt arrangement used for attachment of the port cap.

Performance characteristics of hydraulic motors are well known and defined by relatively simple equations. The most important characteristics relative to EHPM units are related to flow and torque, both being proportional to motor displacement. Flow is also proportional to speed, and torque is proportional to pressure across the motor. Although flow is essentially constant at any given speed, the torque is determined by load and pressure across the motor and is controlled by valves. The maximum motor torque is essentially constant throughout its rated speed range at constant pressure. However, at higher speed, lower pressures are available at the motor due to line and valve pressure drops. This discussion is based on theoretical values that should be adjusted by efficiency. Hydraulic motor efficiency is high when operating at rated pressure and speed. Efficiency is only critical when operating at peak loads. In this area, the overall efficiency is above 85% and the torque efficiency is above 90%. At other than peak loads, the control valve easily adjusts to maintain the desired speed and load torque.

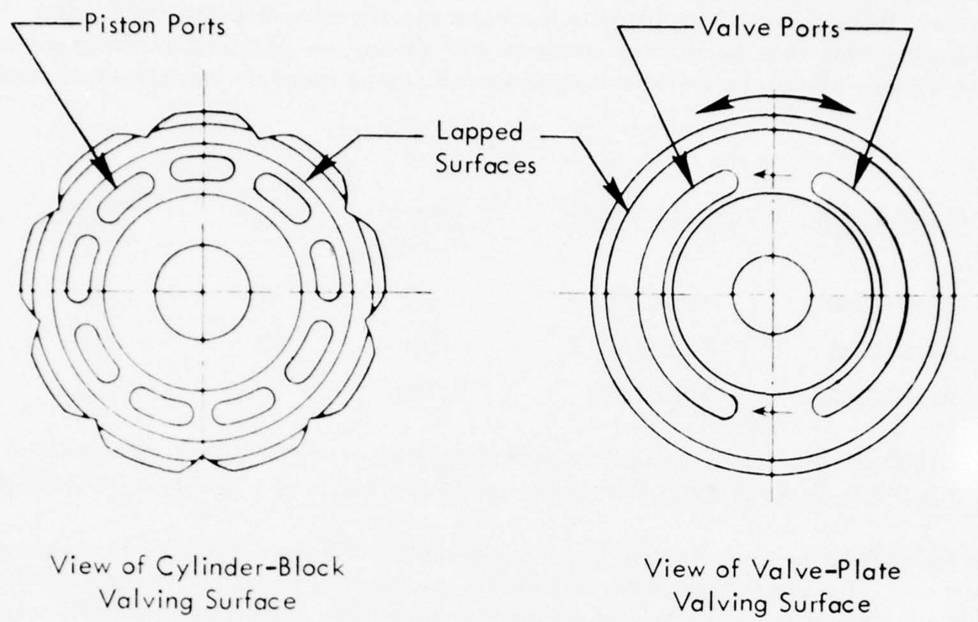


Figure 11 Typical Valving Surfaces Hydraulic Motor

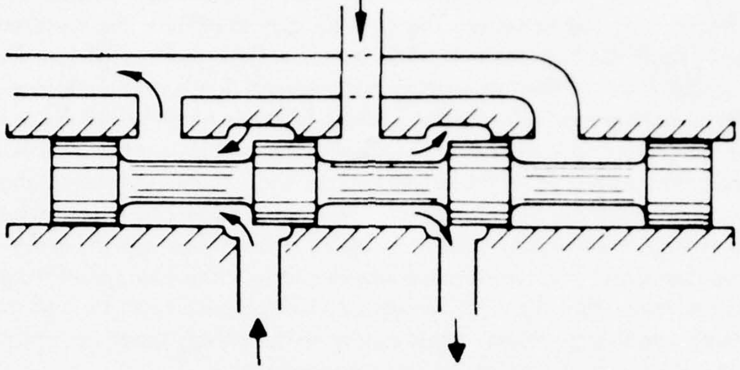
The performance of standard in-line hydraulic motors has discontinuities which are caused by rapid changes in internal leakage at break-out and substantial drop in output torque at low speeds. The in-line motors utilize the metered hydraulic flow to supply both the hydro-static shoe balance flow and the piston displacement (output motion) flow. When operating with normal loads and moderate speeds, they operate smoothly and efficiently. When there is no required output motion or where output velocity commanded is low, operation can be erratic. This is due to the variable flow going to normal leakage, to shoe balance, and to the piston displacement which gives the output motion. For those load conditions where the output torque is higher statically than it is for relatively low speed, jerky operation can be expected until the commanded speed exceeds the low speed range. To avoid this undesirable operation, minimum operating speed must be maintained above 10% of normal operating speeds. This minimum operating speed is compatible with the majority of utility actuation system requirements.

Analysis has shown that unless the driven load is predominantly inertial (such as radar antenna positioning) and low speed operation is required, standard in-line hydraulic motors may be used for utility actuation systems. Most utility applications have both inertia and torque load requirements due to weight and aerodynamic loads. In addition, general operating requirements of utility actuation systems involve movement from one position to another in a specified time and do not involve small corrections.

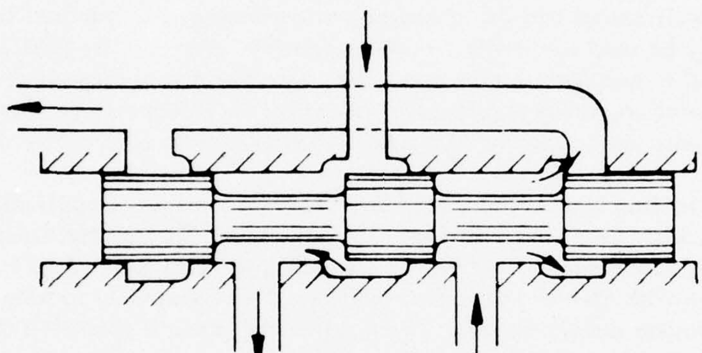
Modified in-line hydraulic motors have been built for servo application of pure inertial loads. One such unit is an ABEX hydraulic servomotor designed to supply the piston leakage and shoe balance flow directly from the hydraulic pressure source rather than with control valve metered flow, thus essentially forcing all control flow to provide output motion. The result is that smooth operation can be maintained to considerably lower speeds. With such a motor, the amount of valve motion required to realize a given static torque is reduced compared to the case where the valve must also supply leakage and shoe balance flows. The output stiffness is, therefore, increased.

<u>Control Valve</u> - The design of modulating control valves is a rather extensive subject covering many types and many construction techniques. To review the various types of valves is not the intent of this study. The most commonly used modulating control valve configuration is the spool and sleeve and it has been selected for the EHPM unit. This type of valve was selected because of the simplicity of its characteristic equations and the general knowledge of construction techniques. In the aviation industry, these valves are used for controlling flow and pressure to flight control actuators, in electrohydraulic valves, and in utility system direction control valves.

The specific spool valve selected for study is a balanced, four-land valve with zero lap as shown in Figure 12. In normal usage, the valve is translated by mechanical or hydraulic means to obtain the modulation of hydraulic pressure and flow. In the EHPM unit, the pulse motor rotates the spool (or the nut) causing translation through the feedback thread or translator attached to the hydraulic motor. The



Four-Land Valve, Balanced



Three-Land Valve, Balanced

Figure 12 Balanced Spool Valve Configurations

following discussion will determine the details of the physical and performance interfaces necessary to integrate the control valve into the EHPM.

Valve Sizing and Flows - In the previous discussion of hydraulic motors, a wide range of available sizes and operating speeds were shown. The motor size (displacement) and operating speeds determine the flow required for motor actuators. The equation for determining motor flow requirements is:

Flow = Displacement x speed/231. When flow is in gallons/min, displacement is in in.^{3}/revolution and speed is in rev/min. The wide range of motor displacement and operating speeds require valve sizes ranging from 1 to 100 gpm as shown in Figure 13.

It is not practical to design a valve for each motor size due to the large variety of motor sizes. A group of standard valves may be designed to match motor sizes and operating speeds.

Valves are sized by flow requirements and pressure drops for minimum physical size and weight. Past efforts to standardize valve sizes have been related to plumbing line sizes and size designation does not always indicate the flow capacity of the valve.

Raymond Atchley, a Division of ABEX, has established electrohydraulic servo valve sizes of 1, 5, 10, 25, and 50 gpm with rated flow pressure drop equivalent to one third of the source pressure. It is interesting to determine the basic dimensions of a series of valves based upon flows and fluid velocities that control the pressure drop. This relationship is stated in the equation:

$$Q = \frac{VA}{.3208}$$

Where Q = flow, gals/min
V = fluid velocity, ft/sec
A = flow area, in^2

Changing A to $\frac{\pi}{4} D^2$ and solving for D (The valve spool diameter)

$$D = \sqrt{\frac{Q}{2.448V}}$$

Assuming a constant velocity of 26.14 ft/sec, a reasonable velocity to minimize pressure drop, the denominator becomes $\sqrt{64}$ or 8; therefore:

$$D = \frac{\sqrt{Q}}{8} \quad \text{at } V = 26.14 \text{ ft/sec}$$

Assuming a series of valve sizes where the spool diameter varies in one eighth inch increments - Q or flow sizes are 1, 4, 9, 16, 25, 36, 49, etc. closely paralleling the Raymond Atchley sizes and providing a series of valves with constant flow velocities. The potential standard size valves are shown on Figure 13. for matching valves and motors.

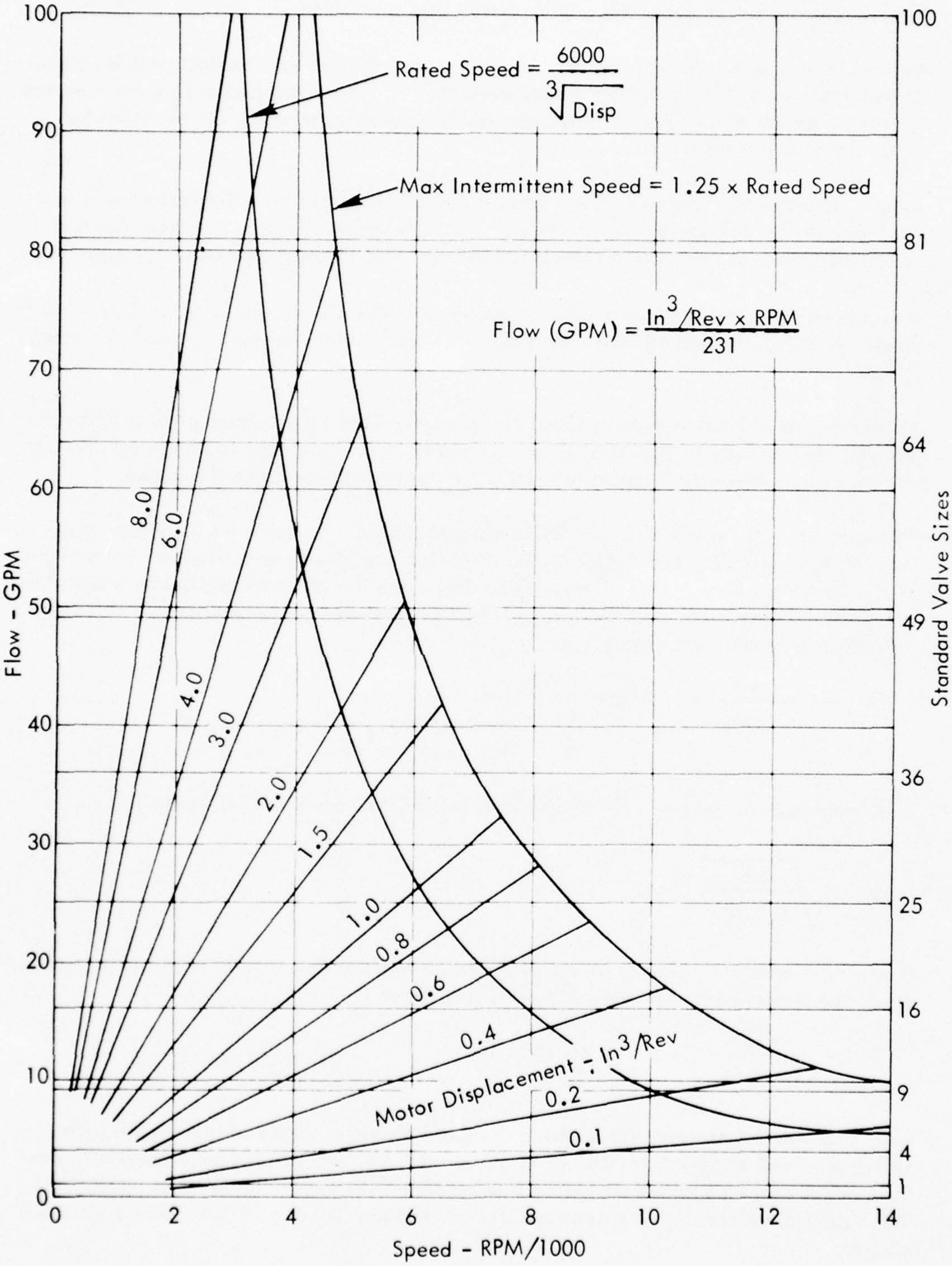


Figure 13 Motor/Valve Matching

Other basic dimensions of the valves can now be determined. The flow areas of the pressure and return ports through the spool should also be of low velocity and are $A = .3208\ Q/V = .01\ (in.^2)$ at a velocity of $32.08\ ft/sec^2$. The stem of the piston should be half of the piston diameter and the outside of the sleeve two times the piston diameter to provide sufficient stiffness and rigidity so as to prevent distortion from pressure and valve housing loads. The land width is also approximately half of the land diameter and the maximum stroke is half the land width to provide sufficiently low valve gain, as discussed later under the stability analysis. One final item prior to tabulating the valve dimensions is the thread of the valve translator or spline which is logically placed on the valve stem, and therefore, is basically half the spool diameter. These dimensions facilitate the use of standard tools and seals. The maximum diameter of the sleeve for all sizes of valves and motors allows axial mounting of the valve within the normal flow paths of the motor port plate.

The standard valve sizes and dimensions of Figure 14. provide minimum pressure drop at rated flows for all areas except the control ports. The two center lands of the four land valve control the flow to and from the hydraulic motor. The design of the ports determine the valve gain and are generally of rectangular shape for constancy of valve gain. Both the inlet and outlet ports are identical and are treated as two orifices in series with the load. Maximum gain occurs at zero load torque and in a 3,000 psi system each orifice has a pressure drop of 1,500 psi. As load is applied, the valve translates to increase the orifice size and provide the required load flow and pressure. It can be shown by calculation that the maximum power a valve can deliver occurs when the valve pressure drop is one third of source pressure. This method of sizing the valve and control orifices is helpful in standardizing valve designs and is often stated as 1,000 psi pressure drop at rated flow for a 3,000 psi source pressure. Caution must be used when source pressures are reduced by line loses at high flow rates or low temperature operation. Under these conditions, the required load pressures may not be available and the valve control orifices may need to be increased in size to decrease drop at high flow rates.

Valve Drive Loads - The predominant forces required to control a spool valve are commonly called flow forces and are directly proportional to the rate of flow through the valve. Valve flow is controlled by the valve lands moving across the metering orifices. The axial force on the spool is equal to the axial component of the net change of momentum. The following equations are derived in Section 10.3 of Reference 3:

$$F = QVp \cos\theta$$

Where:

Q = flow - in^3/sec

V = velocity - in/sec

p = mass density - $lb\text{-}sec^2/in^4$

F = force - lbs

The angle θ is the jet angle of the flow stream leaving the control orifice and is a function of the valve displacement divided by the radial clearance. With a valve clearance of .0001 inch and valve displacement of .006 inch or more, θ approaches 69°. The maximum force occurs at no load when there is a maximum pressure drop across the orifice. In a 3,000 psi system, the maximum pressure drop for each orifice is 1,500 psi. The equation supplies the axial flow force for a single orifice and the force always tends to close the valve.

Flow GPM	Dia. Spool Inches	Dia. Stem Inches	Dia. Sleeve Inches	Width Land Inches	Stroke Inches
1	.125	.062	.250	.062	.031
4	.250	.125	.500	.125	.063
9	.375	.1875	.750	.1875	.0934
16	.500	.250	1.000	.250	.125
25	.625	.3125	1.250	.3125	.156
36	.750	.375	1.500	.375	.188
49	.875	.4375	1.750	.4375	.219
64	1.000	.500	2.000	.500	.250
81	1.125	.5625	2.250	.5625	.281
100	1.250	.625	2.500	.625	.313

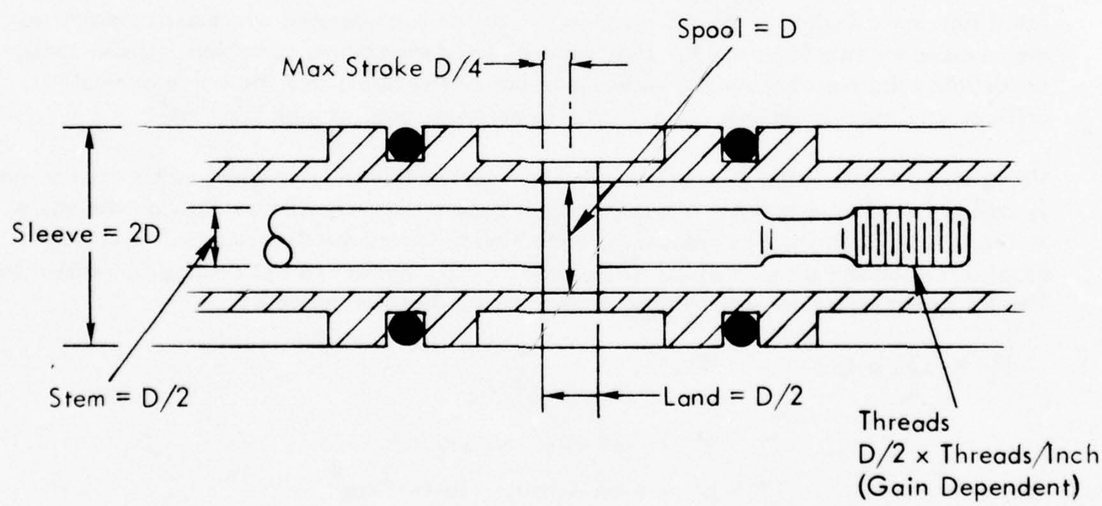


Figure 14 Valve Basic Dimensions

In an actual four-way valve, there are two identical orifices in series, therefore, there is twice the force on the piston. Total force on the piston is:

$$F_T = 2F = 2QV\,p\,\mathrm{Cos}\,\theta$$

Substituting $V = \sqrt{(2\,\Delta\,P)/\,p}$, $\theta = 69^{o}$ and a specific gravity of .85 for MIL-F-5606 fluid provides a further simplification.

$$F_T = 0.0064\ Q\ \sqrt{P_v}$$

where $Q = in^3/sec$, P_v is in psi and is the total valve pressure drop for two identical orifices in series or:

$$F_T = 0.0246\ Q_1\,\sqrt{P_v}$$

where Q_1 = gallons/minute

The total axial forces for valves from 0 - 100 gpm capacity is shown in Figure 15. The maximum force is at no load for a valve P_v of 3,000 psi. Other lines indicate a reduction of flow forces at lower valve pressure drops due to reduced source pressure at the valve or increased load pressure.

The above analysis is made for steady state flow conditions. Under dynamic conditions when the valve and flow must be accelerated, other forces act on the valve. Analysis of the dynamic loads of a typical 25 gpm valve indicate that the loads, with response requirements for utility actuation systems, is less than one percent of the steady state flow forces. The dynamic loads do not occur at peak steady state flow. Therefore, at any other flow sufficient forces are available to accelerate the spool. It is considered appropriate to design for steady state flow forces and ignore the dynamic forces.

Translator Design - In the proposed EHPM, the valve axial forces are converted to torque through a nut and screw arrangement called a valve translator. The screw is attached to the valve stem and the nut is driven by the pulse motor. Three thread standards were evaluated for the translator design: the general purpose 29° acme; the American standard course thread series; and the American standard fine thread series. All of these have standard major diameters to match the valve stem diameters except in the two smallest size valves.

The torque to drive the spool can be determined from the following equation taken from Chapter 5 of Reference 4:

$$T = r_t F \frac{\mathrm{Cos}\ \theta\ \tan\alpha + \mu}{\mathrm{Cos}\ \theta - \mu\tan\alpha}$$

where:

- T = torque - in/lbs
- r_t = thread radius at pitch line - inches
- θ = one half thread angle - degrees
- α = helix angle computed at the pitch line
- μ = coefficient of friction

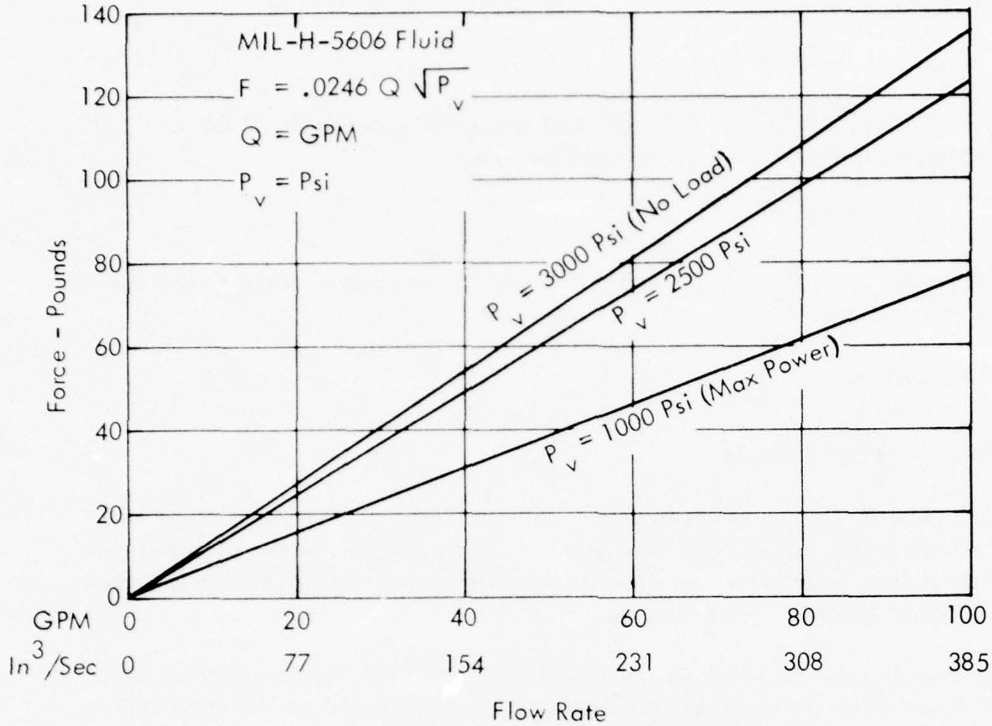


Figure 15. Valve Flow Forces

This equation assumes the use of anti-friction bearings to support the collar loads as in the proposed design.

Figure 16 lists the various valve sizes and data to determine the valve driving torques for both the acme and course thread series. The coefficient of friction is assumed to be 0.05 for a well-lubricated hardened surface. The values of torque versus flow are shown for valve sizes from 0 to 100 gpm in Figure 17. The torque is plotted as a continuous curve to indicate the approximate maximum torque for various size valves at rated flows and maximum pressure drop at no load. For any particular valve, the torque is directly proportional to the flow and the flow forces. In the two thread series analyzed, the torque values are close on all sizes and vary slightly due to thread pitch. The fine thread series was briefly evaluated but considered to be too fine a thread for a power screw application. Non-standard threads were also considered and may be used to provide specific valve gain characteristics, however, in the prototype design standard thread series are used because of available standard tooling.

The translator may be placed at either end of the valve spool. In the Fujitsu unit, the translator is placed at the hydraulic motor end of the spool and the torque required to rotate the spool must be supplied by the pulse motor because the pulse motor drives the spool. At the lower speeds and controlled temperature environment of the machine tool applications, the rotational torque requirements are acceptable. With the higher motor speeds and cold temperature requirements for aircraft applications, the rotational torque loads increase. Because of this increased load, the translator was moved to the pulse motor end of the spool and the hydraulic motor drives the spool through a spline that allows spool translation.

The torque required to rotate the spool is caused by viscous drag forces between the rotating spool lands and the stationary valve sleeve. Newton's equation for determining the relationship between shearing stress in the oil film and the force required can be adapted to the journal bearing. If the speed and viscosity are high and the load is very light, so that the journal is in a central position in the bearing, the following equation, known as Petroff's equation, can be used:

$$F = \frac{\mu v A}{c}$$

Where F is the tangential force, μ is absolute or dynamic viscosity, v is surface velocity, A is journal area, and c is the radial clearance.

Changing the terms and units to a more usable form, the equation becomes:

$$F = \frac{\mu \pi^2 d^2 L N}{c (60 \times 144)}$$

Where d (land diameter), L (total land length), and c are in inches, N (spool speed) in rev/min and μ is absolute viscosity.

GENERAL PURPOSE 29° ACME

$\theta = 14\text{-}1/2°$

Valve Flow Size-GPM	Thread Size	Pitch Diameter Inches	α Degrees	Force Pounds	Torque in Ounces
16	1/4 x 16	.2187	5.2	21.56	5.39
25	5/16 x 14	.2768	4.7	33.68	10.10
36	3/8 x 12	.3333	4.55	48.51	16.97
49	7/16 x 12	.3960	3.83	66.02	25.09
64	1/2 x 10	.4500	4.05	86.23	37.94
81	9/16 x 10	.5125	3.55	109.13	50.96
100	5/8 x 8	.5625	4.05	134.74	74.11

AMERICAN STANDARD COURSE THREAD

$\theta = 30°$

Valve Flow Size-GPM	Thread Size	Pitch Diameter Inches	α Degrees	Force Pounds	Torque in Ounces
4	10 x 24	.1629	4.65	5.39	0.97
9	12 x 24	.1889	4.02	12.13	2.30
16	1/4 x 20	.2175	4.19	21.56	4.96
25	5/16 x 18	.2764	3.66	33.68	9.09
36	3/8 x 16	.3344	3.41	48.51	15.52
49	7/16 x 14	.3911	3.33	66.02	23.77
64	1/2 x 13	.4500	3.11	86.23	34.49
81	9/16 x 12	.5084	2.99	109.13	49.11
100	5/8 x 11	.5660	2.93	134.74	66.02

Figure 16 Valve Torque Calculations

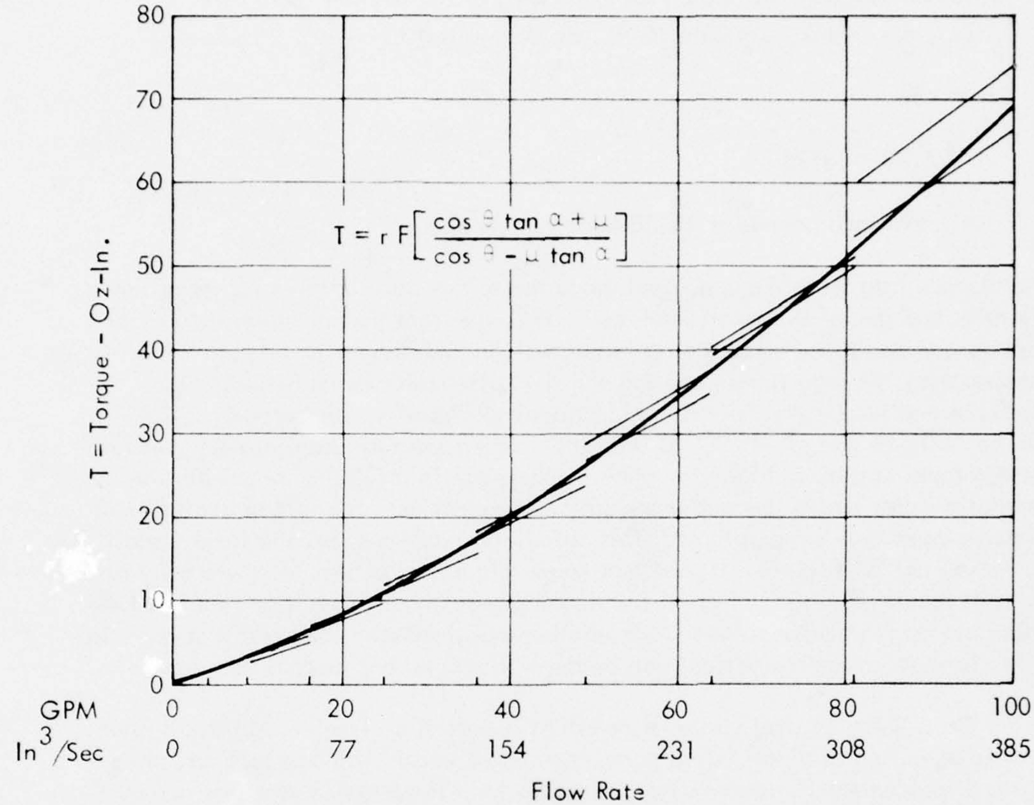


Figure 17 Valve Torque

For a typical 25 gpm, four land valve, running at 4,000 rpm ($d = .625$, $L = .3125$, $c = .0002$, $\mu = 20 \times 10^{-5}$), the tangential force is 2.23 pounds.

$$T = Fr$$

$$= 2.23 \times .3125$$

$$= .696 \text{ inch pounds or } 11.15 \text{ inch ounces}$$

The viscous drag torques are equivalent to the torque due to valve forces at the viscosity and rpm of the valve analyzed. It is apparent that at higher fluid viscosities and rpm's, the viscous drag torque will increase significantly. At low temperature, the drag torque can exceed the pulse motor capabilities. Since the force and torque are directly proportional to viscosity, the torque at -20°F can be 13 times and at -40°F, 30 times normal temperature drag torque. Because these torques can be so high, the change was made to drive the spool with the hydraulic motor where the torque required represents only 5% torque available at minimum operating temperatures. This calculation assumes that the fluid viscosity at the journal bearing stays at ambient temperature viscosities. The viscosity may be reduced locally due to friction of the rotating spool, but this action requires time. Other design features such as increased clearance, reduced land width or grooving of the land (a common practice) can be done to reduce the viscous drag forces.

Valve Gain - The control valve in an EHPM is part of a closed loop servo system. Prior to selection of detail valve parameters (flow versus displacement and pressure versus displacement characteristics), a closed loop servo system analysis is required. This analysis determines the highest closed loop again that can safely be used in the system and satisfy the stability criterion.

The closed loop servo system within the EHPM includes the following components: the hydraulic motor; the control valve; and the feedback translator device. Stability criterion for this type of system have not been published, but good design practice dictates a gain margin of at least 6 db; i.e., the loop gain could be increased by a factor of two before encountering system instability. This criterion is consistent with MIL-F-9490D which specifies gain and phase margins as a function of airspeed and aeroelastic mode frequency. In general, worst case stability for an EHPM system would occur at zero airspeed. For this condition, MIL-F-9490D specifies a gain margin of 6 db with no phase requirement below minimum operational airspeed.

Derivation of the expression for the closed loop gain of an EHPM operating at zero airspeed (no load) is presented in the design section for the prototype EHPM. The following equation may be used for any system after the loop gain and stability criterion has been developed:

$$\text{Loop Gain} \quad \left(\frac{1}{\text{Sec}}\right) = \frac{Kv\ Kt}{D}$$

Where: Kv is the control valve flow gain at null (in^3/sec-in), Kt is the translator feedback gain (in/rev), and D is the hydraulic motor displacement (in^3/rev).

A simplification of this equation can be made if the flow to be determined at one revolution of the spool is desired. The number of threads per inch drops out and the loop gain is expressed in terms of flow and motor displacement as follows:

$$\text{Loop Gain} \quad \left(\frac{1}{\text{Sec}}\right) = \frac{Q\ (in^3/sec)}{D\ (in^3/rev)}$$

It is emphasized that the loop gain must be determined at no load flow which is the maximum valve gain and worst case stability. Assuming that the valve has one third of the source pressure drop at full load, the no load flow is the ratio of $\sqrt{Ps/Pl}$ or 1.732 Q where Q is the valve flow at maximum EHPM operating speed.

Open Versus Closed Loop Trade Study

Traditional technology to interface low level electrical command signals with a drive system is to use an electrohydraulic servo valve in a closed loop system. Feedback signal to the servo valve is usually taken from a position sensor on the load. Special cases exist where a mechanical feedback is used in lieu of electrical feedback and others where a velocity transducer feedback signal is used in combination with the position sensor feedback. These are used in a small percentage of applications. A block diagram of a closed loop electrohydraulic servo system is shown in Figure 18.

This technology, however, has not been employed to drive aircraft utility systems. The primary reason is the lower total cost of a simple mechanical input system and the inherent reliability of mechanical controls. With the advent of low cost, high reliability digital control technology, however, the use of servo systems to control aircraft utility functions seems to be feasible. The advantages of such an arrangement include the capability to reduce overall system weight, reduce cost, introduce modifying controls not feasible with the mechanical input systems, and reduce the exposure of the hydraulic system to fatigue from pressure surges.

A digital closed loop system of the type shown in Figure 18 was evaluated against an open loop EHPM system shown in Figure 19. The results of this qualitative evaluation were put in matrix format in Figure 20.

It is concluded that the EHPM is well suited for aircraft utility functions. The response requirements (bandpass) of utility systems are much lower than flight control systems where Electro Hydraulic Valve (EHV) servos are extensively used. It is concluded that the bandpass of the EHPM is not a penalty for utility systems.

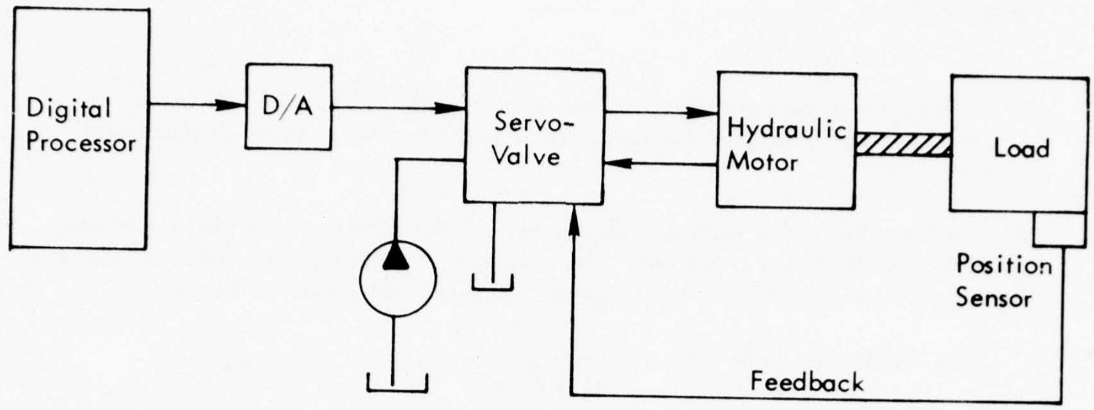


Figure 18 Closed - Loop EH Servo System

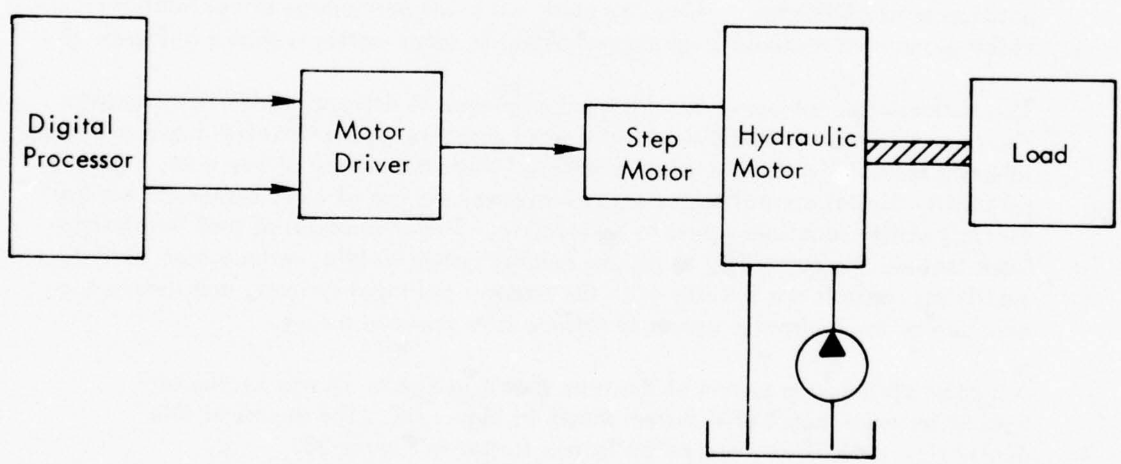


Figure 19 Open - Loop EHPM System

Characteristics	EHV Servo	EHPM
Nature of Device	Displacement Input, Position Output	Pulse Train Input, Position Output
Inputs	Analog Voltage - Other Inputs through D/A Converter	Uniform Pulse Train or Random Positive or Negative Pulses
Outputs	Depends upon Position Feedback Gain	Exactly Determined by Number of Input Pulses
Feedback	Position of Output Member; Sometimes Velocity	Not required
Loop Gain-Linearity	Set High for High Linearity	Stepping Linearity does not require Feedback
Stability	Simple Analysis Usually Sufficient	Same
Error	Depends upon Feedback Transducer Resolution, Linearity, Quadrature Noise, etc.	Depends upon Accuracy of Individual Stepping Mechanism Within Limits of Designed Stepping Rate
Bandpass	Zero to High Frequency Response Device	Relatively Low Frequency Response Device
Digital Programming	May be Adapted with D/A Input Device and Data Encoder on Output Shaft	May be Commanded Directly by Computer or Tape Pulse Train
Quiescent Power (Typical Application)	.2 Watts Electrical; 455 Watts (.34 gpm) Hydraulic	12 Watts Electrical; 320 Watts (.24 gpm) Hydraulic
Life-Reliability	Dependent upon Amplifiers, Feedback Transducer, Hydraulic System Cleanliness, D/A Converter	Dependent upon Shaft Bearing, Thermal Environment Drive Circuits, Other Mechanical Elements of a Single Motor
Cost	Dependent upon Cost of Amplifier, EH Valve, Feedback Transducer, D/A Converter, Valve Manifold, Provisions	Dependent upon Cost of Step Motor, Drive Circuits, Valve Manifold, Mounting Provisions
Weight	Usually Less than 1 Pound for EHV	High Slew Speed. Stepper Motor Weight is about 7 Pounds (present state-of-the-art)

Figure 20 Closed-Loop EHV Servo Versus Open-Loop EHPM

Open loop operation of an EHPM requires safeguards against loss of pulses. If the motor misses an input pulse, system positioning accuracy is impaired. The magnitude of the error is directly proportional to the number of pulses lost.

It is postulated that because of improved reliability of electronic circuits, that loss of pulses will be a remote occurrence if the system is operating within its design regime and it appears feasible to shift regimes during operation to keep the system within an operable range. It also is feasible to shut the system down if an operable regime is exited.

Using Reference 5 as a guide to the closed loop control of stepper motors, it is concluded that open loop control with fault monitoring is the better approach. Closed loop control, however, does allow higher speeds and provides more positive positioning integrity. The complexities introduced, however, detract from its appeal. For example, closed loop control requires a feedback transducer, usually an optical encoder disk, which supplies the necessary information to close the loop. To determine the proper rotor positions at which phase switchings should occur requires either a fixed encoder slot/motor detent position angle or the introduction of a time delay circuit in the feedback path. Dual sensors are required for direction sensing. Finally, speed of the closed loop servo is quite sensitive to load variations. Since the motor load varies directly as a function of speed, the closed loop control would involve additional complexity to control slewing speeds. These disadvantages led to the exploration of fault monitoring schemes for improving positioning integrity. The open loop system was examined to determine criteria for pulse positioning integrity. In order to avoid loss of pulses, the following criteria must be met.

- Stepper motor torque must be sufficient to drive valve under all operating conditions.
- Hydraulic motor flow must be sufficient to provide the commanded motor velocity.
- Hydraulic motor supply pressure must be adequate to supply load torque.
- Motor resonance velocity versus load combinations must be avoided.

It is obvious that mechanical failures within the EHPM or the actuation mechanical system could produce conditions in which the criteria for positioning integrity could not be met. For these eventualities, special monitoring provisions must be supplied. The non-failure states should also be investigated to determine if the above criteria could be met. Areas open to question are: 1) Cold temperature operation, 2) Step motor resonance conditions.

Tests will validate normal operational capabilities of the EHPM system. The rationale for concern at low temperature involves hydraulic fluid viscosity which increases line drops and results in lower available pressures and flows. Motor specification data reflect no resonance zones, and analytical determination of such conditions is of sufficient difficulty to warrant experimental test results to reveal latent resonance zones.

Several industrial users of open loop EHPM's utilize a simple monitoring scheme to automatically shutoff the unit whenever pulse positioning integrity cannot be met. It consists of switch contacts mounted in the gearbox between the step motor and control valve. As the spool is displaced beyond a predetermined overtravel, the switch contacts are closed sending a signal to either shutdown or slow down the input pulse rate.

A straightforward concept such as this one used for industrial controls can be successfully employed in a utility aircraft system such as the flap system. Most failures in the drive system will result in a hardover control valve spool. Using this insight into system operation, switch contact points can be positioned so that contact is made when the valve spool has traveled a predetermined distance beyond the full open position. The making of the switch contacts generates an interrupt signal in the microcomputer which halts the pulse train and provides pilot warning through the existing annunciation system.

To complement this monitoring system, an asymmetry detection system may be employed. It consists of shaft encoders mounted on the flap drive torque tubes in the left and right wing. A critical difference between the left and right shaft encoder results in automatic shutdown and engagement of the drive shaft brakes which lock the flap system in position. This system detects faults in the structure between the drive motor differential and the flap surfaces.

In order to protect the input drive system from failures, redundant input sensors, microcomputers, and power drive circuits can be configured. The prototype system, however, is a single system and is subject to single failure points in the input drive system.

SECTION III
APPLICATION SURVEY

One task of this program is to survey the C-5 or similar aircraft for possible uses of the EHPM concept in utility and secondary flight control systems. To perform this task comprehensively and effectively it is necessary to recognize useful alternatives some of which involve use of new technology and to consider the EHPM concept in its larger scope - i.e. combined with its digital computer. The following items are specifically noted:

- The approach is applicable to the control of linear hydraulic actuators as well as hydraulic motors. One linear actuator device, described in reference 6, consists of a hydraulic actuator with a screw mounted axially through the head end and then through the piston into the rod, arranged so that the screw is rotated by the linear motion of the piston. The rotary motion of the screw is used as the feedback in the same way that the rotary motion of the hydraulic motor is used in the EHPM. The actuator can be called an electrohydraulic pulse actuator, EHPA, to distinguish it from the EHPM.

- A black box which incorporates a digital computer which may be multiply redundant can control one or more subcircuits or parts of subcircuits. This box can control any subcircuit which lies within its capacity range and the only difference between two black boxes controlling different subcircuits is the ROM (read only memory) program. That is, the hardware is identical. It is therefore, predicted that such a black box will be a functional standard and will be used to control different subcircuits in the same airplane and in different airplanes. This broad usage will lead to significantly lower costs than normal for black boxes. This prediction is substantiated by the story of the electronic slide rule.

- Two or more actuators using the same pulse stream for control will operate in synchronization - i.e. their displacements will correspond. This type of control has in the past been difficult to achieve and has usually resulted in a mechanical tie between actuators. When using the EHPM approach with an encoder the encoder outputs are compared and can make adjustments or stop the system if the actuators exit synchronization.

- Adding control parameters to a circuit increases the complexity of the system only to the extent of the input signal, e.g., a pressure transducer and its wiring. The circuit to multiplex, digitize, and read many inputs is assumed to exist in the standard black box.

- Adding an EHPM or EHPA to the system involves adding the actuator, its driver, an encoder if used, and interface functions which are standardizable.

Lockheed's cargo aircraft were surveyed to determine where the EHPM/EHPA concept would be a viable alternative if the aircraft were being designed within the next decade. This survey revealed the following candidate applications:

SUBSYSTEM	IS EHPM/EHPA A VIABLE CANDIDATE?		
	C-5	C-141	C-130
Thrust Reverser	YES	NO	NA
Landing Gear and Door Actuation	YES	YES	YES
Nose Gear Steering	YES	YES	YES
Kneeling System	YES	NA	NA
Cross Wind Gear Positioning	YES	NA	NA
Aft ramp actuators	YES	NO	NO
Visor Actuator	YES	NA	NA
Flaps/Slats	YES	YES	YES
Winch Control*	YES	YES	YES
Petal Doors	YES	YES	NA
Stabilizer	YES	YES	NA
Stabilizer Trim	YES	YES	NA

* When used for special mission aircraft - not on all aircraft

Additional candidate applications which are used in other aircraft are as follows:

Radar drive
Wing fold
Wing sweep
Bomb bay doors
Gun drives

Comparative Analysis

In evaluating the viability of the concept for each of these applications it is necessary, to evaluate the following factors:

- Procurement Cost
- Maintenance Cost
- Weight
- Reliability

Safety is not listed since it is assumed that adequate safety is built in before the above are considered.

As it is not feasible to perform quantitative analyses for this generalized evaluation a discussion of the factors is presented.

Procurement Cost - the concept transfers the bulk of the control functions from the hydraulic circuit to the electronic circuit. The hydraulic valving complexity is significantly reduced. The electronic circuitry is standardized in easily replaceable modules. The number of LRU's (line replaceable units) in the system is significantly reduced and the ones that are required can be identical for many subcircuits. The effects of standardization - larger quantities of fewer pieces will result in lower costs to specify, procure, test, develop, store, and install.

Maintenance Cost - the simpler hydraulic circuit and the modularized electronic circuit coupled to the multiply redundant computer which can self test to identify faulty modules offers a real opportunity to reduce maintenance costs. Fewer parts leads to reduced logistics costs.

Weight - weight for most of the applications will be less than the present approach primarily because of less valving and plumbing.

Reliability - The number of moving parts in the systems is dramatically reduced, and the degree of redundancy in the control is increased. Many limit switches may be eliminated.

It is predicted that dramatic improvement in the maintenance and reliability factors and significant improvement in the procurement cost and weight factors can result.

SECTION IV
PROTOTYPE EHPM DESIGN

Selected Actuation System

The existing flap control and actuation system involves a complex of mechanical, hydraulic, and electrical devices and was designed and arranged using the available technology at the time the C-5 was developed. A hypothesis leading to this study is that dramatic developments in electronic circuitry especially in the computer field can lead to significantly improved control and actuation systems. An approach being considered is to accumulate as much of the control as possible within a digital computer and simplify to a maximum the hydraulic valve system and the electrical controls. The C-5 flap and slat actuation system offers an attractive arrangement for making a comparative study and for evaluating a prototype system. The hydraulic control is sufficiently complex that substantial simplification can be demonstrated. Further justification for this selection is provided by the availability and adaptability of the C-5 Iron Bird for prototype testing. The full size simulator provides actuation hardware including gearboxes, torque tubes, actuators, and load simulation equipment.

The elements of the C-5 flap system are indicated in Figure 21. The torque requirements for the hydraulic motor are shown in Figure 22. The various speed and gear ratios used in the system are provided in Figure 23.

Description of a Projected System - The elements of a system to do the same job as the C-5 flap and slat system using current and projected technology is shown in Figure 24 which also indicates the prototype test system. In this arrangement, the crew input is a dual electronic signal from the flap control handle. Each of the two drive packages is controlled by an electronic package which is a digital computer plus associated electrical circuitry related to the power delivered to the stepper motor drive. The electronic control reads the various input devices which are the crew input, the motor output, the gearbox output, the asymmetry detectors and hydraulic pressures, and on the basis of these inputs, controls a stream of electrical pulses to the stepper motor that drives the hydraulic motors. The electronic control also delivers signals to the hydraulic insolation valves and to the asymmetry brakes on the basis of the inputs which it has read.

Much of the circuitry in the electronic control is projected to be form and function standard. It will not be circuitry which is specifically developed for this special task. It consists of a standardized CPU, ROM, and RAM plus some specialized circuitry. (CPU = Central Processor Unit; ROM = Read Only Memory; RAM = Random Access Memory)

It is the ROM circuitry which adapts the standard hardware to the specialized task of controlling the flap system. The ROM hardware is standard but its numerous elements are programmed -- positioned permanently or semi-permanently. The ROM provides the information which defines how the CPU will handle the information it reads from the various inputs and from the RAM. The RAM is variable memory and is temporary in nature.

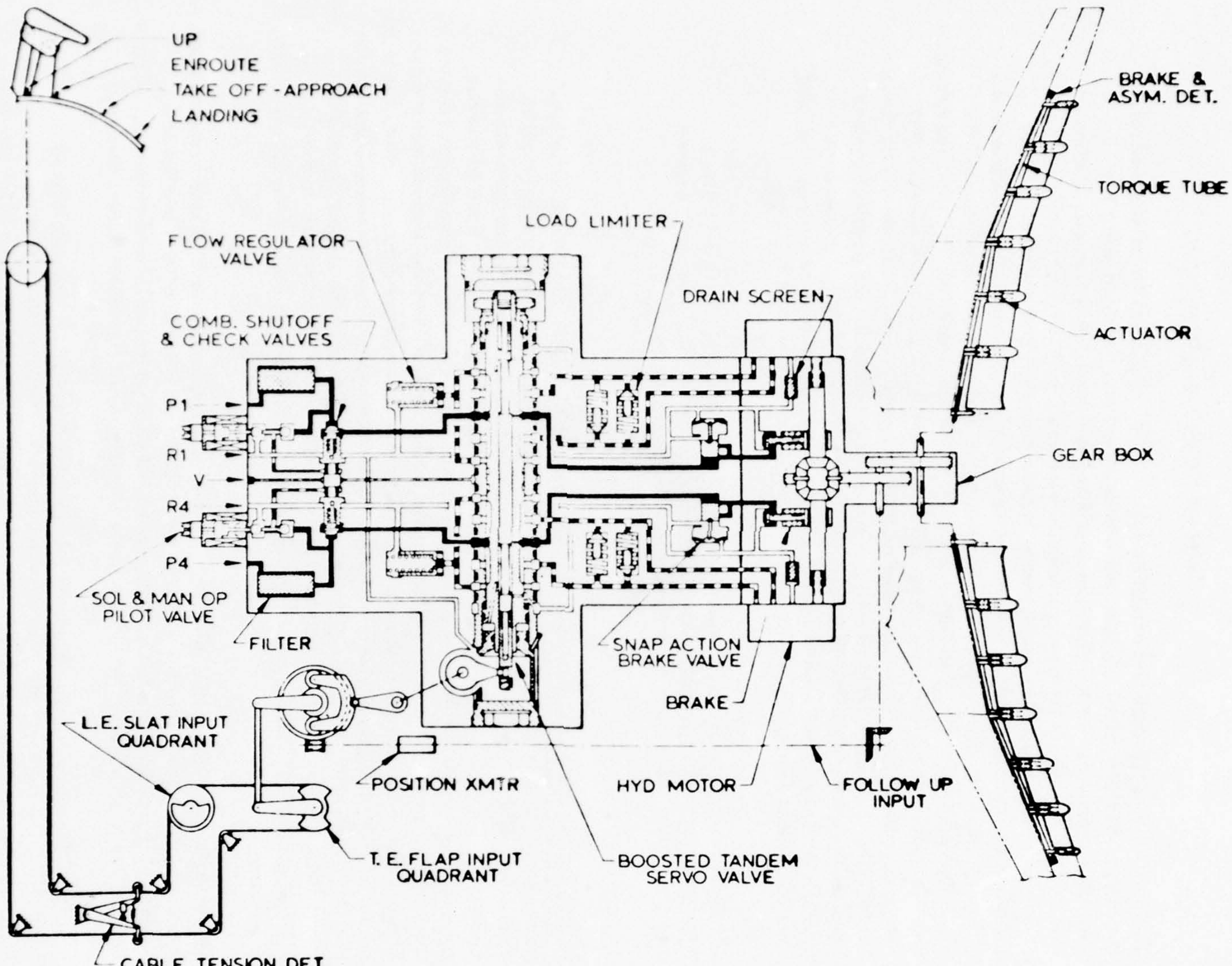
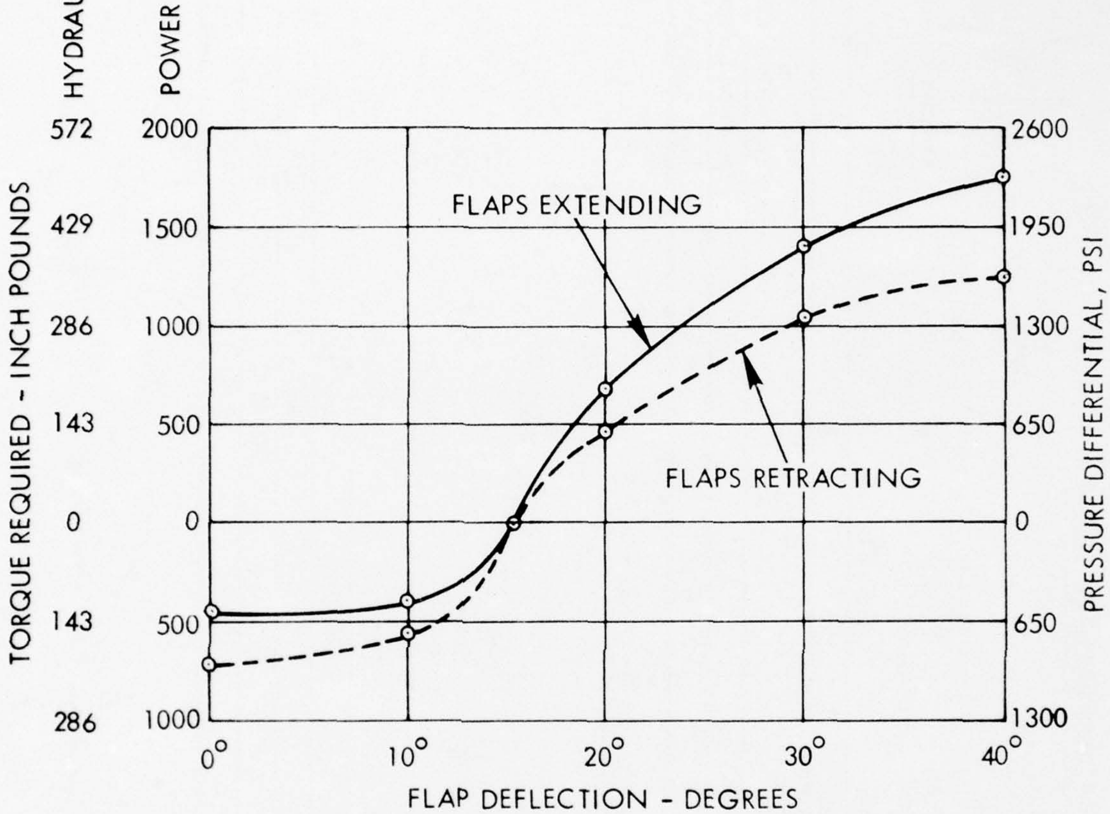


Figure 21 C-5 Trailing Edge Flap Schematic



NOTE: PRESSURE DIFFERENTIAL ACROSS 1.52 IN^3/REV MOTOR WITH OVERALL EFFICIENCY OF 0.9 FOR MOTOR/GEAR BOX PACKAGE TO PROVIDE TORQUE INDICATED. MOTOR TO POWER PACKAGE GEAR RATIO 3.5/1.

Figure 22 Flap Load Vs Deflection

POWER PACKAGE DATA

Shaft	Revolutions Required to Extend	Speed (RPM)	Ratio to Output Shaft	Remarks
Output Shaft	526.19	1127.64	1:1	
Hydraulic Motors	1841.665	3939.74	3.5:1	
Worm Shaft	21.925	45.8	24:1	Shaft that drives position comparison shaft
Position Comparison Shaft	315.72°	1.83	600:1	Position Comparison shaft
Input Lever	103.50°	0.602	1830:1	Follow Up Mechanism

ACTUATOR ASSEMBLY DATA

Shaft	Input Revolutions Required to Extend	Input Speed (RPM)	Ratio Input to Output	Output Speed (RPM)	Output Revolutions Required to Extend
No. 1 Tee Box	526.19	1127.64	1.0589:1	1064.92	498.29
Step Gearbox	498.29	1066.92	1.875:1	514.626	265.043
2 Through 4 Gearboxes	526.19	1127.64	1.9853:1	514.626	265.043
5 and 6 Gearboxes	526.19	1127.64	2.4545:1	460.	214.377
7 Through 12 Gearboxes	526.19	1127.64	2.68181:1	420.	196.207

Extend Time: $28.0 \, {}^{+0}_{-5}$ Seconds

Figure 23 Flap System Data

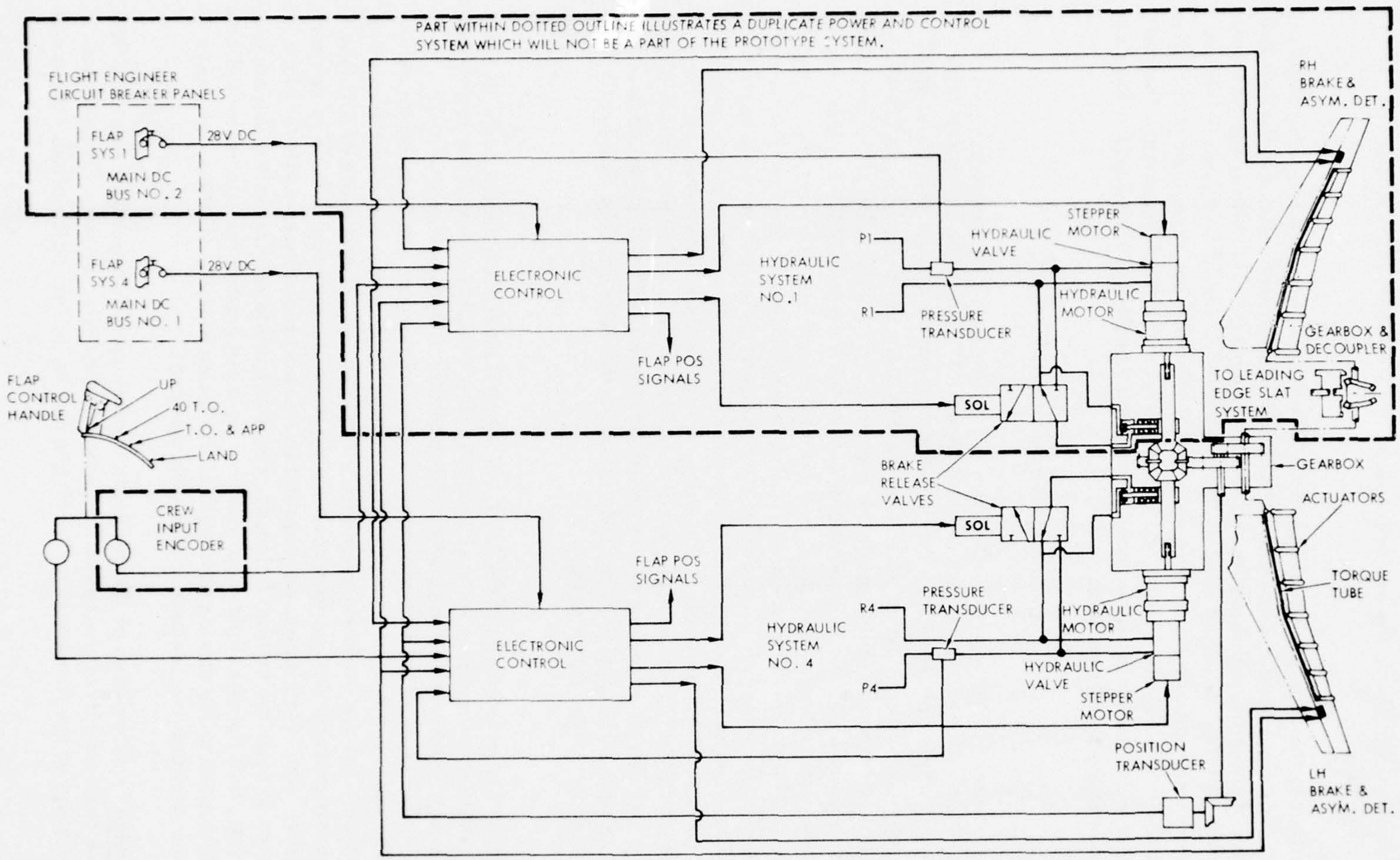


Figure 24 System Arrangement for Prototype Testing

The fact that the electronic control can be produced in a small size, at low cost, and with high reliability is the key technology advancement which makes this approach feasible.

Modification to the flap drive package to facilitate using only the left wing flaps is required. The right hand hydraulic motor and brake are deactivated and the left and right gears of the differential are locked to the same shaft. This allows the left motor to drive the differential output gear at the same speed it would be driven if left and right motors were operating and since only 1/2 of the flaps are connected to the drive torque tube, the left motor will carry its normal 1/2 of the total flap load. The test drive motor will, therefore, be exposed to the same conditions as if would if the complete system were being operated.

Motor/Valve Assembly

Motor Selection - *The present C-5A flaps are driven by Vickers bent-axis* 3915-30 motors. These motors have a displacement of 1.52 cubic inches per revolution, and a single unit will drive all the wing flap panels on one wing. An equal displacement axial piston motor is selected for the design of the prototype EHPM unit. The axial piston design is selected for the simplicity and compactness of the complete motor/valve assembly. The selected motor is the ABEX model AM8C-2. This unit is presently in production for the Boeing 747 on the inboard flap drive system. A similar model, with slightly less displacement, is also in production for the F-111 wing sweep. The model AM8C motor is fully qualified for both commercial and military applications. An assembly drawing of the AM8C-2 is shown in Figure 25. Performance curves for this unit are shown in Figure 26. Referring to the flap load requirements of the previous section the highest pressure required at the motor is 2,362 psi. Since identical size motors drive the present and proposed prototype EHPM system, no problem related to torque sufficiency is expected. Analysis indicates that the flap application does not require a servo motor design.

In the design of the prototype EHPM unit, no internal modifications are made to the standard AM8C-2 unit except for seal changes. The design features of the motor/valve interfaces can be seen in the complete EHPM assembly drawing, Figure 27. To adapt the valve housing to the standard motor, the normal port cap is removed and is replaced by the valve housing. A port plate is used to interface between the rotating cylinder block and the stationary aluminum valve housing. A port plate of this type is used in many pump or motor designs to reduce weight when the port cap is large or contains additional valves. No changes are required in this port plate/motor cylinder block porting area to alter the critical balance. The spool spline driving unit is driven through a radial slot, that permits slight misalignment, by a pin pressed into the motor cylinder block. The spool spline driving unit utilizes the inside diameter of the spool sleeve as a bearing thus maintaining concentricities with the spool.

Valve Design - The design of the prototype control valve is dependent upon the flow and torque requirements of the hydraulic motor and flap drive system. The maximum operating speed of the flap system is 3,949 rpm and for further calculations, the speed

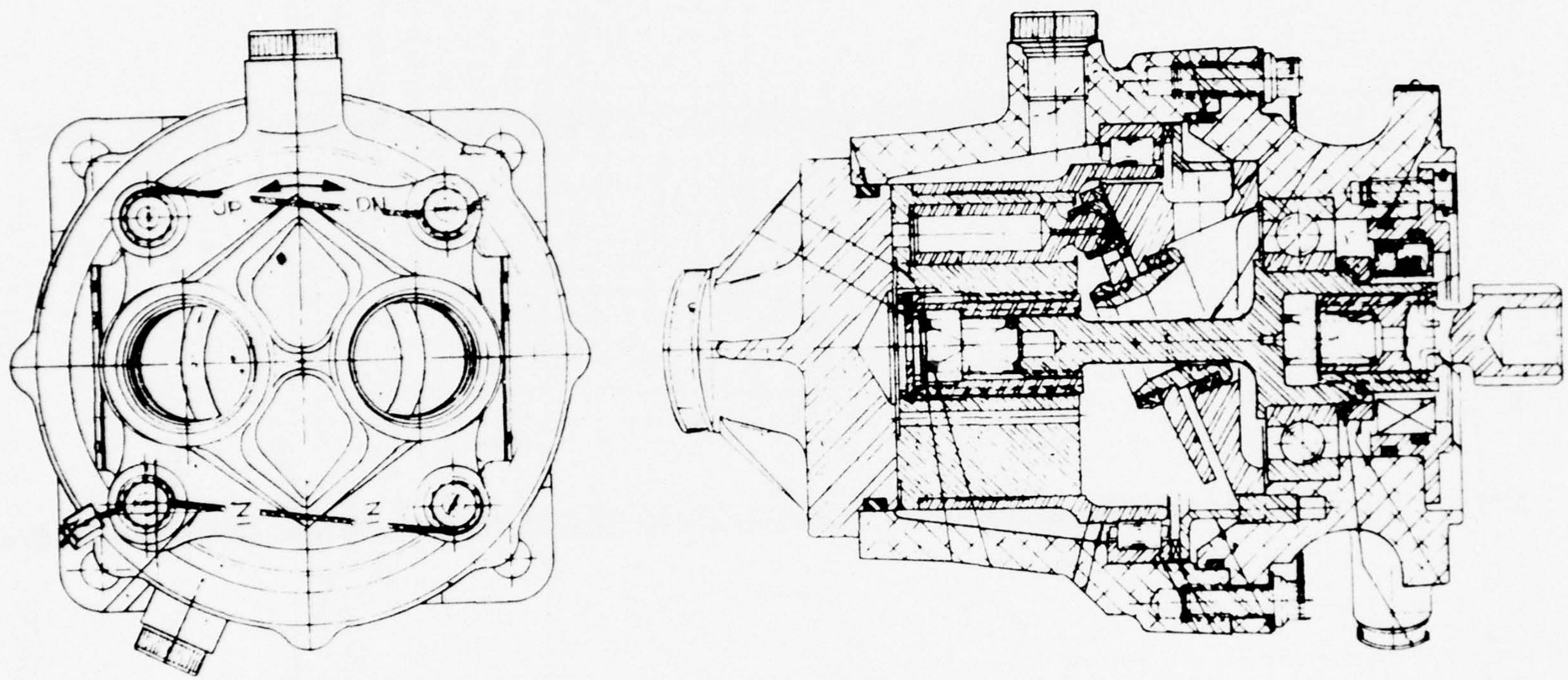
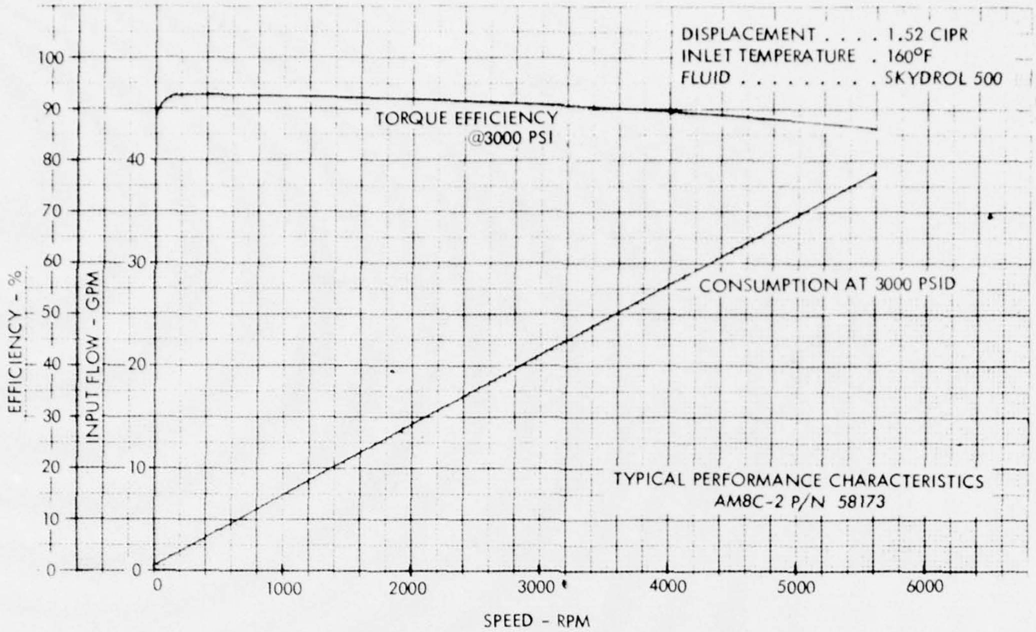


Figure 25 Hydraulic Motor, Abex Model AM8C-2



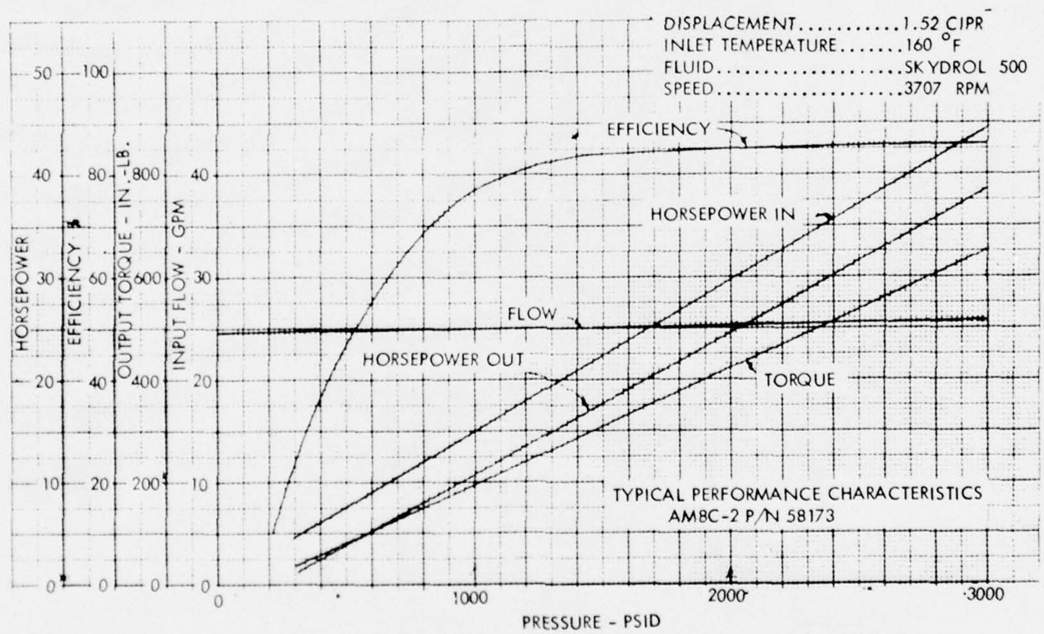


Figure 26 Hydraulic Motor Performance

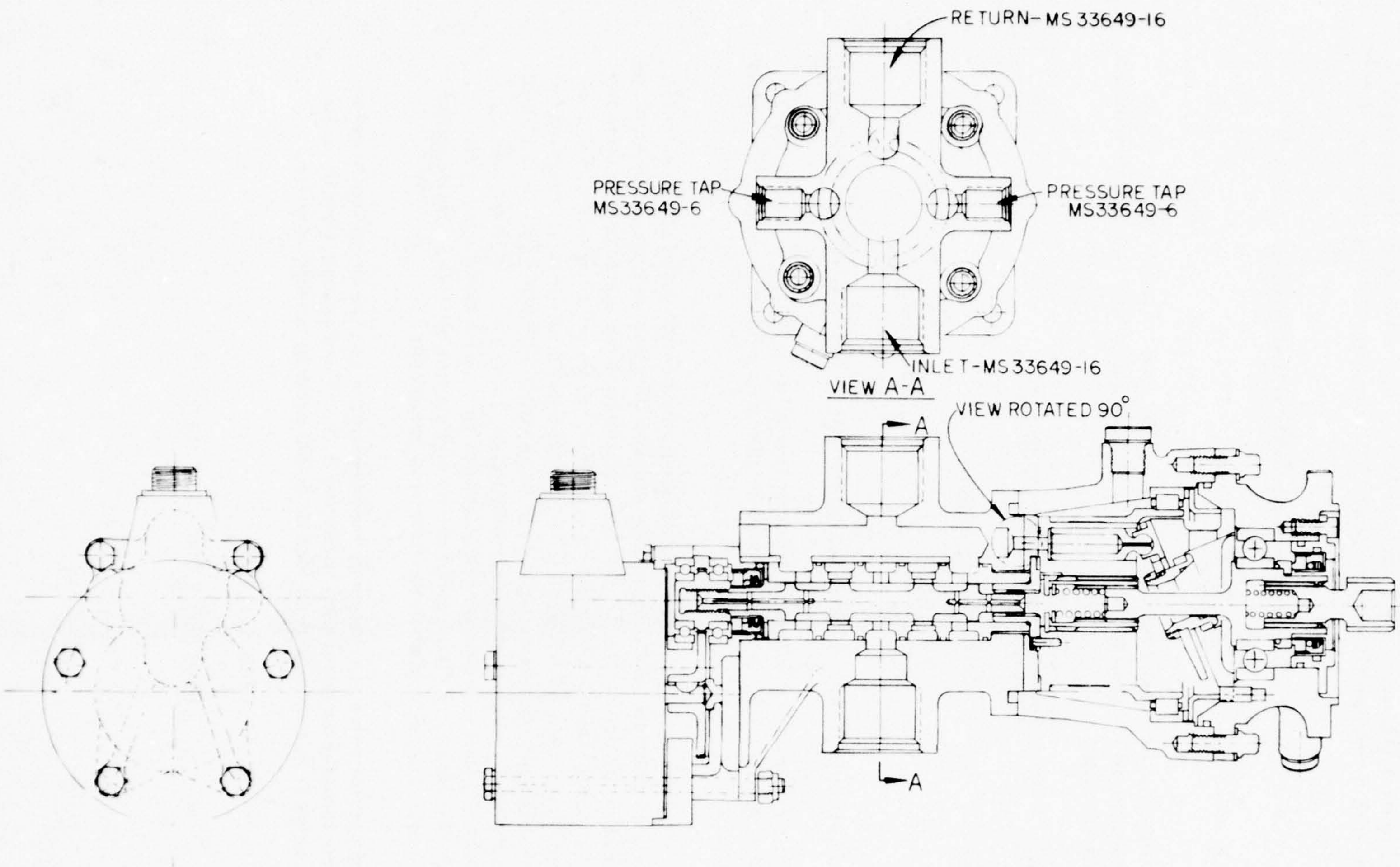


Figure 27 Electrohydraulic Pulse Motor

is considered to be 4,000 rpm. The selected AM8C-2 hydraulic motor displacement is 1.52 in^3/rev with a volumentric efficiency of .97. The control valve maximum flow is:

$$Q = \text{Displacement} \times \text{speed}/(231 \times \text{efficiency})$$

$$\text{or } Q = \frac{1.52 \times 4{,}000}{231 \times .97} = 27.1$$

The motor output torque requirement of the flap drive system is shown in the previous section. The pressure required at the motor to develop the torque can be determined by the following equation:

$$P\ (\text{psi}) = \frac{2\ \pi\ T\ (\text{in/lb})}{\text{Displacement } (\text{in}^3/\text{rev}) \times \text{efficiency}}$$

The maximum pressure requirement of the motor is:

$$P\ (\text{psi}) = \frac{2\ \pi \times 500}{1.52 \times .9} = 2{,}296 \text{ psi}$$

This maximum pressure is only required during the last 5° to 8° of flap extension.

At all other times, the required motor pressures are below 2,000 psi. In the last 5° to 8° of flap extension, the motor drive may be slow to provide the required increased load pressure.

With the flow and torque requirements established, the control valve size is selected from the group of standard valve sizes. The standard 25 gpm valve size is selected for the basic valve dimensions. The 27 gpm flow requirement represents an 8% increase in flow with slightly higher valve pressure drop that can be adjusted in the design of the control orifices. An ABEX Model 425 electrohydraulic servo-valve designed for a rated flow of 25 gpm incorporates a second stage valve with basic dimensions almost identical to the 25 gpm valve standard developed in this study. This Model 425 valve was selected for the prototype EHPM control valve with modifications for incorporating the translator drive, spool drive, and control orifices. The design of these modifications are discussed in the following paragraphs.

The selected control valve is a four-way balanced valve and the equations to determine the flow forces were developed in Section 3.0 . The maximum flow force for the 27 gpm valve is determined at no load or 3,000 psi drop across the valve.

$$F = .0246\ Q.\ \Delta\ P_v$$

$$= .0246\ (27)\ 3{,}000 = 36.38 \text{ lbs}$$

This valve flow force is transferred to torque through the selected thread and gearing of the translator. The standard valve translator thread size can be either a 5/16 x 14 Acme or a 5/16 x 18 course thread. The 5/16 x 14 Acme thread was selected for the prototype design. The torque can be determined by the equation developed in the section on translator design. The torque requirements for a standard 25 gpm valve is 10.1 inch ounces and since the flow forces and torques are directly proportional to flow, the torque requirement of the prototype valve of 27 gpm is increased by a ratio of 27/25 x 10.1 or approximately 11 inch ounces.

Valve Stroke Versus Flow Selection - As determined by the stability analysis which follows, a valve gain of 40 should not be exceeded. This requirement pertains to operation near the shutoff regime. After the valve is open to some degree, it is allowable to increase the gain. The selected stroke versus flow requirement for valve manufacture is shown in Figure 28. For this figure, the Δ P across the total valve is held constant at 3,000 psi until design flow is obtained and then the flow is held constant while the valve is opened to its maximum. Differential pressure, of course, has to be reduced as the valve is opened so as to maintain constant flow. At valve full open, the total Δ P across the valve is 500 psi. As determined previously in this section, the required ΔP for the hydraulic motor itself is 2,296. At valve full open, total ΔP across valve and motor is 2,796 leaving about 200 psi for loss in the hydraulic system.

Servo System Stability Analysis - The servo valve, hydraulic motor, and mechanical feedback linkage form a closed loop servo system. The stepper motor is located outside of the loop and, therefore, its response characteristics are not a part of this analysis. However, since it is the servo valve driver, its response (to accelerate and decelerate) and its maximum speed characteristics should equal or better that of the rest of the servo system.

The stability criterion used for this analysis is a gain margin of 6 db. Military specification MIL-F-9490 calls for a gain margin of 6 db and phase margin of 30° Since there are no active feedback sensors in the system, it is considered that the phase margin can be neglected. The analysis consists of root locus and frequency response characteristics for a linearized math model of the servo system.

Using the following parameters for the EHPM prototype system, an analysis of the unloaded motor stability is performed. This is comparable to bench operation of the EHPM and is a worst case stability consideration.

V	=	Oil Volume	=	2.5 in.^{3}
J	=	Hydraulic motor inertia	=	0.0082 lb. -in. -sec.^{2}
B	=	Adiabatic bulk modulus of oil	=	0.27×10^6 lb./in.^{2}
D	=	Hydraulic motor displacement	=	1.52 in.^{3}/rev. = 0.242 in.^{3}/rad.
L	=	Total leakage coefficient	=	1.0×10^{-3}(in.^{3}/sec.)/(lb./in.^{2})

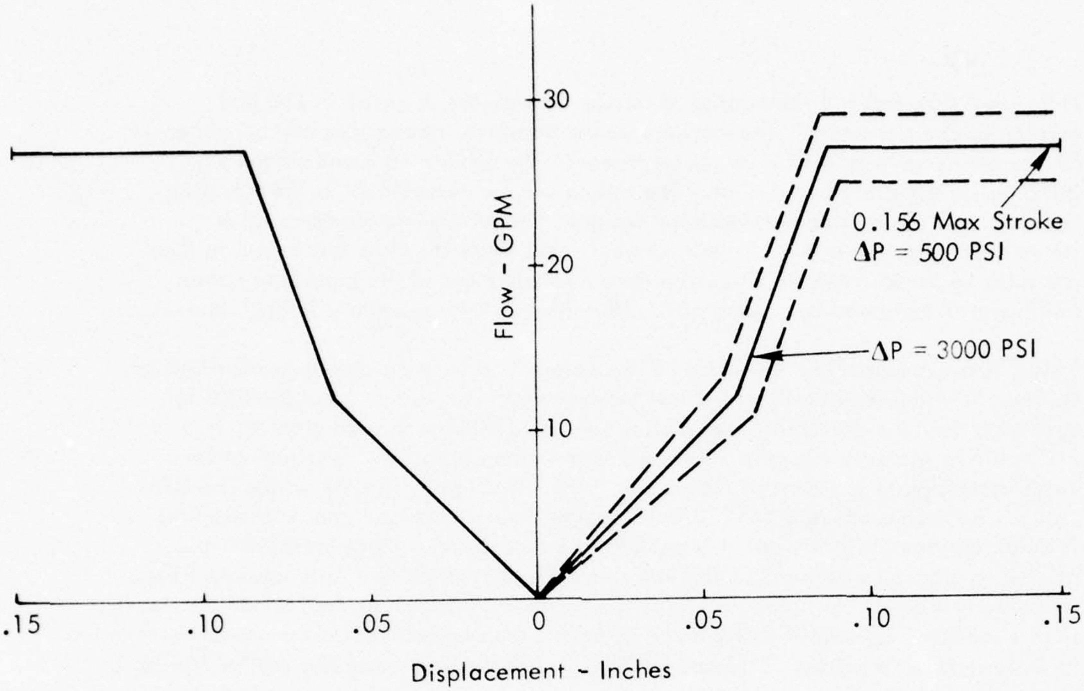


Figure 28 Valve Flow vs Displacement

Root loci were calculated for a variation of loop gain, K_{loop} from 1 to 100 sec $^{-1}$. The servosystem is quite stable for the entire range of gain variation and for bench operation with the minimal inertia indicated. Natural frequency for this condition is high (approx. 650 rad./sec.) with damping inversely proportional to loop gain.

The effect of adding load inertia is examined.

Choosing a first cut value of loop gain of 40 sec. $^{-1}$, the load inertia is varied from 0.02 to .09 lb-in-sec.^{2}. This corresponds to ground operation of the system for increasing inertia loads. The result of this analysis is seen on Figure 29 (Plot No. 1). In general, the effect of adding load inertia is a reduction of natural frequency and a slight decrease in damping.

The load inertia of the C-5 trailing edge flaps reflected to the hydraulic motor shaft is estimated to be .064 lb.-in-sec^2. The stability margins of the loaded servo-system are determined. First, a root locus is determined for the servosystem with a conservative load inertia estimate of 0.07 lb-in-sec.^{2}, varying the closed loop gain constant from 10 to 100 sec.$^{-1}$. From the results, also plotted on Figure 29 (Plot No. 2), it is seen that the system goes unstable for gains somewhat above 100 sec.$^{-1}$. To assure a minimum 6dB gain margin, the loop gain should be held to 1/2 this value or 50 sec.$^{-1}$. To check stability margins, an open loop frequency response was conducted on the model. The results, plotted on Figure 30, show gain margin of 8.68 db and a phase margin of 87 degrees using a gain of 40 sec.$^{-1}$. Therefore, the stability criterion of 6 db gain margin is met.

The effect of hypothetical hinge moment loads was then examined. To simplify the problem, the flap system hinge moment coefficient was considered to be constant at 0.0432 in-lb/rad. A root locus of the inflight configuration of Figure 29 (Plot No. 3) was solved for loop gain variations related to the ground operation gain variations. The results show that the system response, for any load inertia or gain variation, remains essentially unchanged for the air spring loads under consideration here, 0 to 0.15 in-lb./rad.

Stepper Motor and Gearing

Load Analysis and Gearing Requirements - A most important consideration in the selection of a motor is the analysis of the motor load, both running torque and inertial torque. The valve flow torque load was calculated at 11-inch ounces at 27 gpm (4,000 rpm). In addition the friction and inertia loads may double the pulse motor torque requirement.

A review of available EPM's, on the previous "baloney" chart, Figure 2, shows that few manufacturers can meet the speed-torque requirements. Performance data for candidate motors from three manufacturers are shown in Figure 31. Stepper motors were chosen

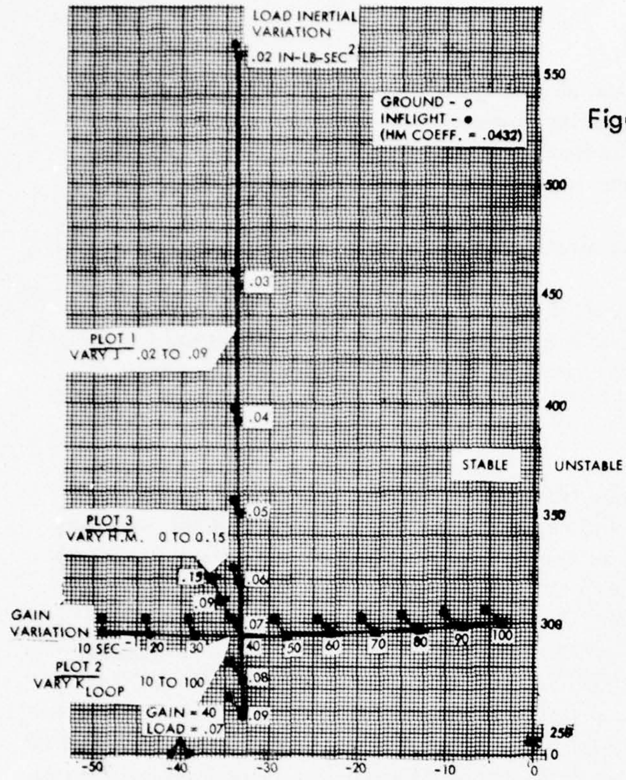


Figure 29 S-Plane Root Loci, Hydraulic Servo Systems

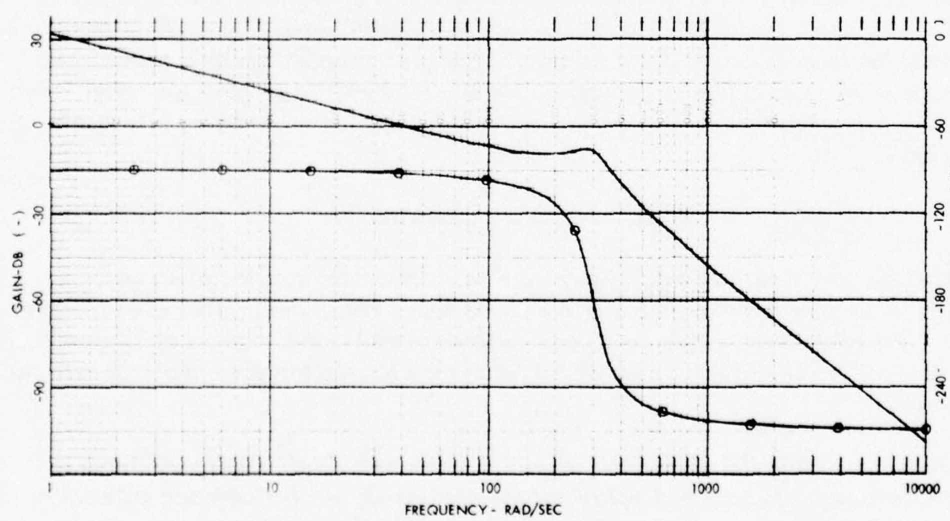


Figure 30 Open Loop Frequency Response

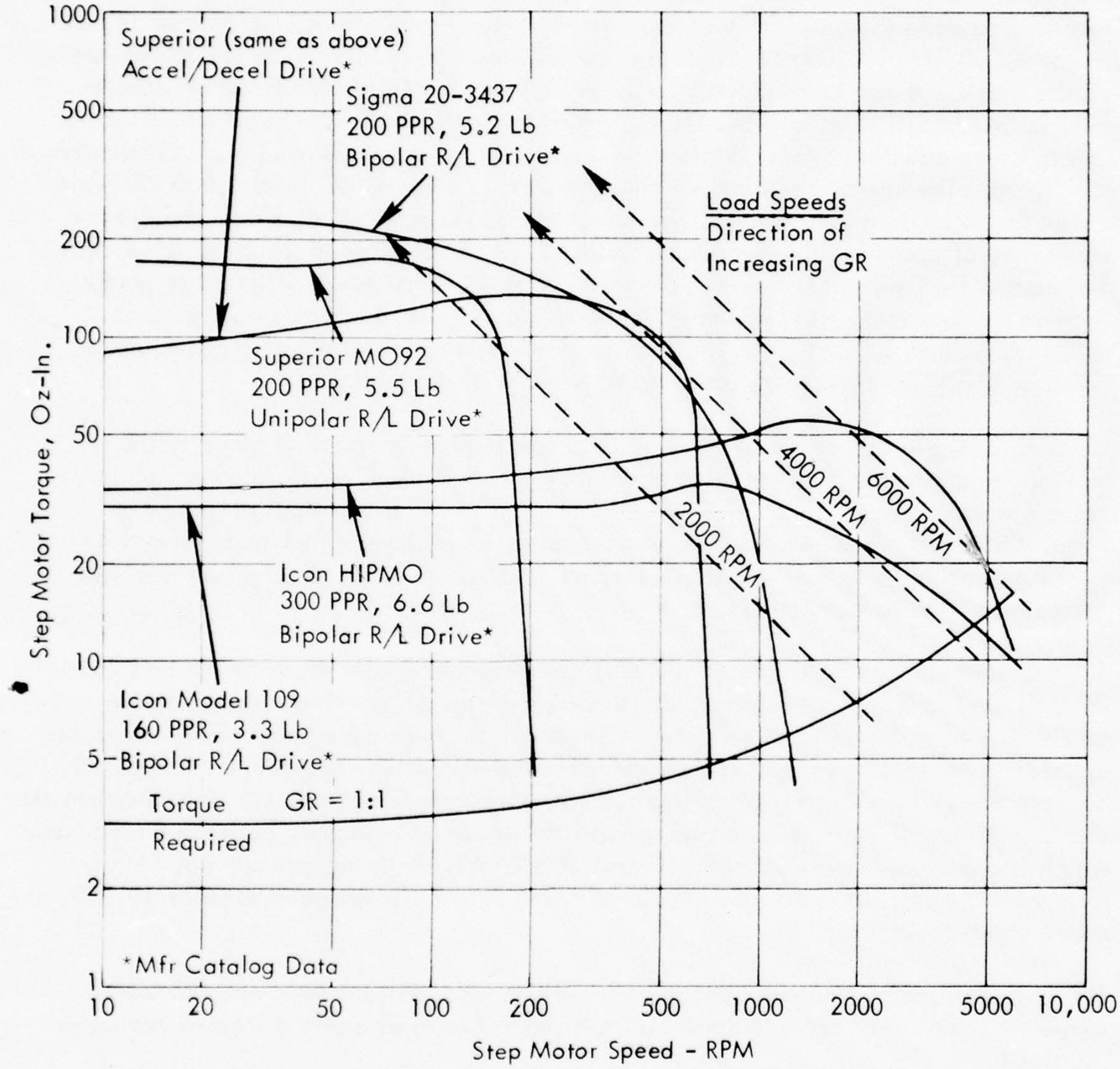


Figure 31 Candidate Step Motor Speed-Torque Characteristics

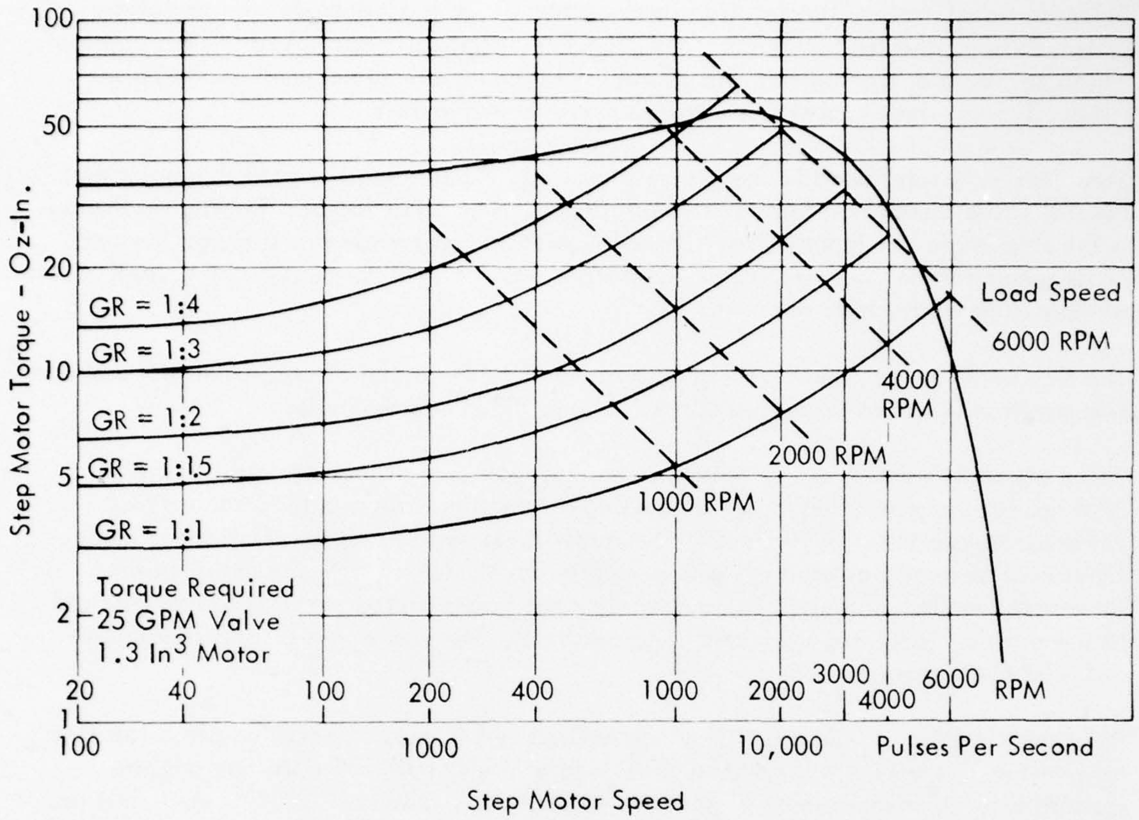
for this figure based upon advertised speed-torque characteristics with an upper limit placed on the weight of the unit of 10 pounds. The figure indicates that without resorting to gearing, only the Icon unit can meet the design point, motor load 11 oz. in. at 4,000 RPM (10,000 PPS). The effect of input drive circuit choice is also noted on the graph. It can be seen that the Superior Electric Motor M092, which is being used in SAAM EHPM units with success, exhibits greatly increased capabilities when driven by an accelerate/decelerate drive unit. The effect of gearing is also shown on the figure. The operating points for three speeds, (2,000, 4,000, and 6,000 RPM) are shown for increasing gear ratios. The effect of increasing gear ratio does present the possibility of operating the Superior M092 unit with accelerate/decelerate drive units. The margin for load estimate error, however, was not considered sufficient to pursue such an arrangement. It is pointed out that the only unit capable of operation at speeds as high as 6,000 RPM is the Icon Hi PMO motor. This unit with gearing can drive an EHPM for overspeeds up to 6,000 RPM for test purposes.

Gearing between the stepper motor and load can reduce the running speed of the motor. In addition to the small losses due to gear inefficiency, gearing does present certain disadvantages. It changes the output shaft rotation per input pulse of the hydraulic motor. Gearing also requires increased motor torque in direct proportion to the gear ratio, and inertia loads reflected back to the motor through the gear are increased by the square of the gear ratio.

The selected stepper motor (Icon Hi PMO) speed-torque capability is shown on Figure 32 with load torque requirements plotted for 5 gearing ratios. The abscissa of the plot is shown in both RPM and pulses per second. This was done because of an imposed upper limit of 10,000 pulses per second input command rate. This limit was imposed because the input pulse train is being generated by the microcomputer. It was estimated that a minimum of 100 μs of computation time would be necessary between each pulse, which is equivalent to an upper limit pulse rate of 10,000 pulses per second. A gear ratio of 1.2 results in rated load speeds of 4,000 RPM for input pulse rates of 10,000 pulses per second.

The torque difference between the motor speed-torque characteristic and the load torque characteristic curve is the torque available to accelerate the load to overcome inertial loads.

The upper limit of load inertia allowable for the Hi PMO motor as published in the motor specification data is 3.5×10^{-4} lb-in-sec^2. The inertia calculations which follow this paragraph show that the stepper motor load inertia expected for the prototype systems is 2.17×10^{-4} in-lb-sec^2 which leaves a margin of 38%. The primary contributor to this value is the two inch diameter gear on the motor shaft which has an inertia of 1.8×10^{-4} in-lb.sec^2. The reflected inertia of the control valve spool is negligible. (Note: After fabrication and test of the prototype EHPM the motor inertia load was recalculated to be 4.5×10^{-4} in-lb-sec^2. The increase was due to the addition of the translator nut and bearing inertia. The increased inertia increased the acceleration time constants experienced in the prototype testing).



Flow Torque - $T_F = 2.2 \times 10^{-3} n$ where n = valve speed in RPM

Shaft Seal - $T_S = 3.12$ oz-in.

Total Load = $(T_F + T_S)/GR$ where GR = Step Motor Speed/Valve Speed

$$= \frac{3.12 + 2.2 \times 10^{-3} n}{GR}$$

Figure 32 Speed-Torque Characteristic, Icon HI PMO Step Motor, 6.6 Lb, 300 PPR

Since an acceleration/deceleration ramping technique is planned for the prototype system to bring the load to rated speed, it is important to investigate the time constants which can be programmed into the microcomputer software to accomplish the pulse ramp. Typical ramp format was shown previously in Figure 8.

Step Size - As mentioned in the previous section, the gearing between the motor and control valve assembly affects system resolution. The other factor affecting resolution is the step angle per input pulse. The EPM selection was made primarily on the basis of its speed-torque characteristic. Resolution was not important, it being several orders of magnitude better than required.

The flap angle is not linear with hydraulic motor position and the maximum slope of the flap position vs pulses delivered curve is about .0004 degrees/pulse.

There are other advantages to having a small stepping angle besides resolution. An EPM has several operating modes depending on stepping rate and load conditions. These modes are the stepping mode, the transitional mode, and the slew-speed mode. Operation of utility aircraft systems is usually accomplished with the motor in the slew-speed mode. The stepping and transitional modes are purposely avoided because of the erratic, jerky motion imparted to the load. The step angle affects the granularity of the response.

Environmental Considerations - The most critical environmental consideration is motor temperature. Motor windings have temperature ranges within which they operate satisfactorily dependent upon class of insulation used. Normally MIL-Spec motors use Class H insulation which allows a rated motor temperature of 130°C. Exceeding this value reduces winding life. Several options are available to provide cooling; namely, encasing the EPM in a cooling shell and blowing with fans, providing fins on the motor, or cooling with hydraulic fluid. During the prototype development program, motor temperature will be monitored and cooling provided if required.

In isolated cases, motors may be used in dirty environments. The "dirt" could be sand or dust, or other contamination detrimental to the motor, such as hydraulic fluid. For these operations, protection of the motor from its environment may be required. Either a protective case or a specially built motor offer a solution to the contamination problem. The EPM selected for the flap drive EHPM system is presently being used in Fujitsu EHPM numerical control application and is provided with dirt and hydraulic fluid contamination protection.

Mechanical Requirements - Principal stepper motor mechanical requirements are mounting considerations.

The physical dimensions of the selected stepper motor are presented in Figure 33.

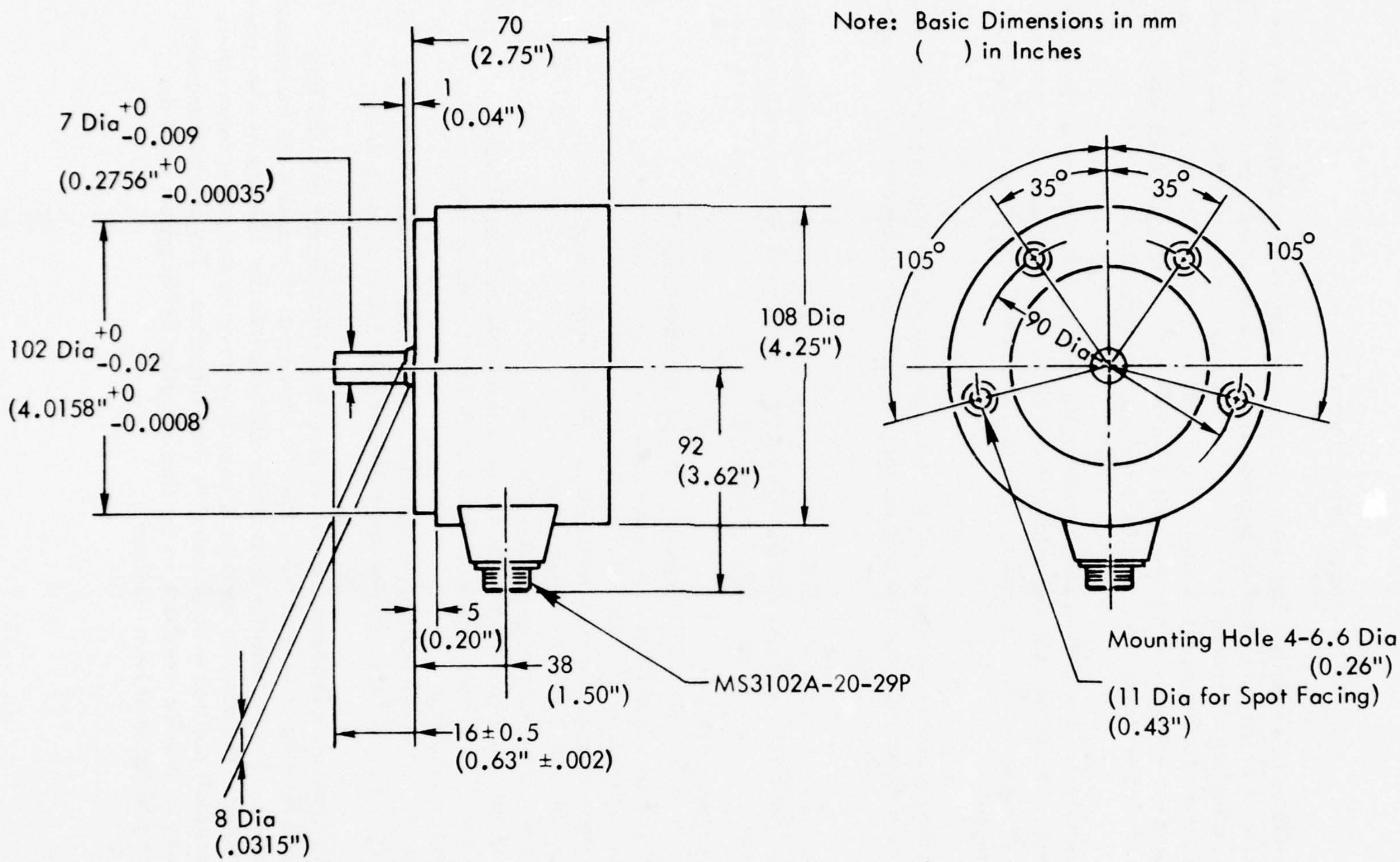


Figure 33 Outline Drawing - HI PMO Stepper Motor

Input Drive System

Stepper Motor Drive Units - The motor windings are sequentially energized by the driver circuit which consists of the sequence logic, power drivers, and limiters. The step and direction commands are input to the sequence logic where they are converted to base signals for the power drivers. The signals are amplified by the power drivers and routed via voltage and current limiters to the motor windings. Additional discussion of stepper motor drive circuits is provided in Appendix A.

For simplicity and low cost, a series resistance driver is chosen for use in the prototype study. The driver selected is Icon Model 601-T, which is described more completely in Appendix A. The packaging of this unit is not representative of aircraft hardware.

In open loop operation acceleration/deceleration controls are required to ensure that pulses are not lost. For the test program these controls are incorporated in the software of the microcomputer.

Microcomputer - An ALTAIR 8800 microcomputer is used to program the pulse train for prototype EHPM testing. The ALTAIR 8800 microcomputer is discussed in detail in Appendix B. A brief summary of the unit is:

Processor: 8 bit parallel
Maximum Memory: 65,000 words, all directly addressable
Instruction Cycle Time: 2 microseconds
Inputs and Outputs: 256, all directly addressable
Number of basic machine instructions: 78 (181 with variants)
Add/Subtract Time: 2 microseconds
Number of subroutine levels: 65,000
Interrupt Structure: 8 hardware vectored levels plus software levels
Number of auxiliary registers: 8 plus stack point, program counter and accumulator
Memory Type: Semiconductor (dynamic or static RAM, ROM, PROM)
Memory Access Time: 850 ns static RAM; 420 or 150 ns dynamic RAM

Feedback Elements - Encoders are used to provide position signals from the flap handle and the flap drive gear box. The two basic types of encoders are incremental and absolute. The incremental encoder is noise sensitive and loses count when power is interrupted. The absolute encoder is insensitive to noise and power interruptions and senses shaft position without losing its final reference. The absolute encoder was chosen to provide feedback in the flap actuation EHPM system since the reference position must be maintained.

SECTION V

PROTOTYPE FABRICATION

As-Built Configuration, Input System

The input system was fabricated by Lockheed with off-the-shelf hardware and does not represent the packaging which will be used for a production system. The adequacy of input system technology to provide an appropriate package - size, weight, and cost - was a key consideration in establishing this program, but the development of production configurations was not an intent of the program. The details of the input system are presented in Appendix B.

As-Built Configuration, EHPM

The prototype EHPM was manufactured by ABEX Corporation, Aerospace Division, Oxnard, California, Part Number 63085, Model Number SMP8C-1. Drawings of the EHPM installation (63085) and assembly (62285) are shown in Figure 34, Sheet 1 and 2 respectively. Sheet 3 identifies major elements of the valve drive and feedback considered to be the key technology items of the EHPM. Data applying to these assemblies are as follows.

- Spool and Sleeve Assembly
  1. Material MIL-S-7420 Cond. E (E 52100).
  2. Valve diametral clearance 0.0003 inches.
  3. Valve travel/gain - see figures 42 and 43.

- Translator Driver
  1. Material - screw (spool) MIL-S-7420 Cond. E (E52100) nut (nut - gear driven) 440C Cond. A.
  2. Thread - 0.3125-14 Class 4G ACME thread.
  3. Spline - 0.3125 pitch diameter, 10 teeth, 32/64 diametral pitch, 30 degree pressure angle.

- Gearing
  1. Material - CRES Per QQ-S-763 Class 440C Cond. A.
  2. Gearing - USA standard 20 degree pressure angle fine pitch involute spur gear, 32 diametral pitch machined to AGMA quality number 12.

- Bearings
  1. Light series angular contact bearings preloaded to 50 lbs. with wavy washer springs.

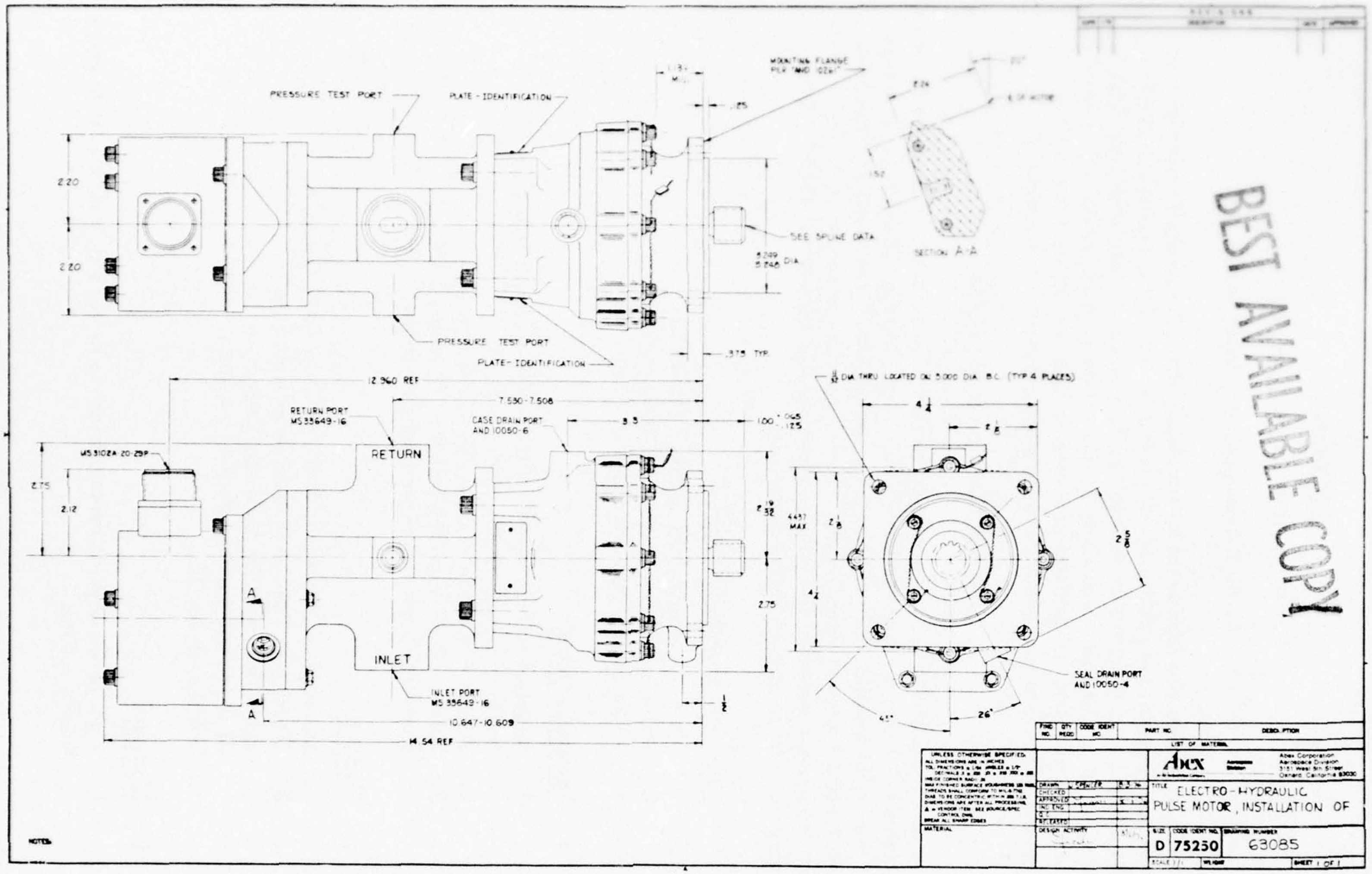


Figure 34 Abex Drawing of EHPM (Sheet 1)

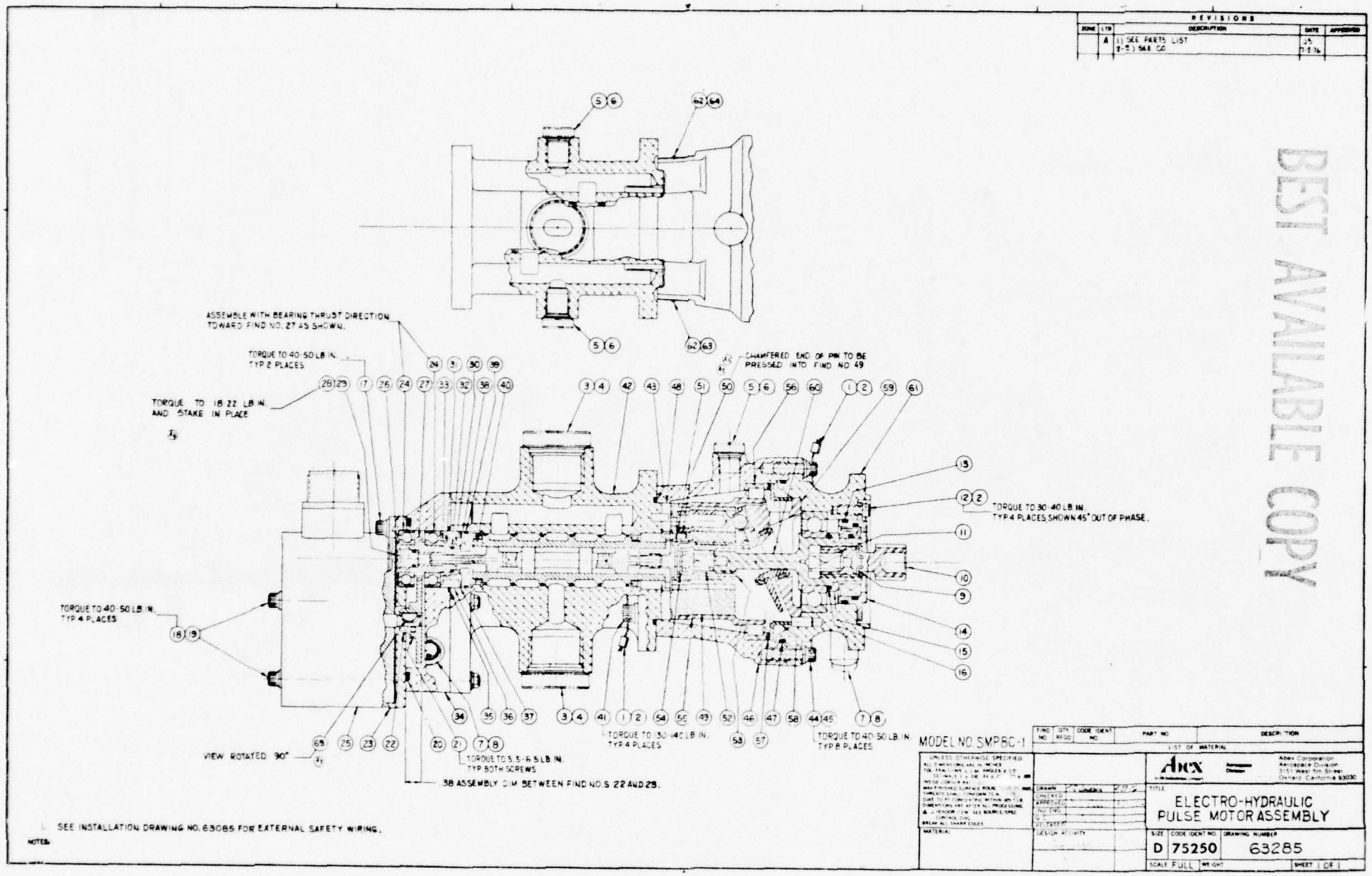


Figure 34 Abex Drawing of EHPM (Sheet 2)

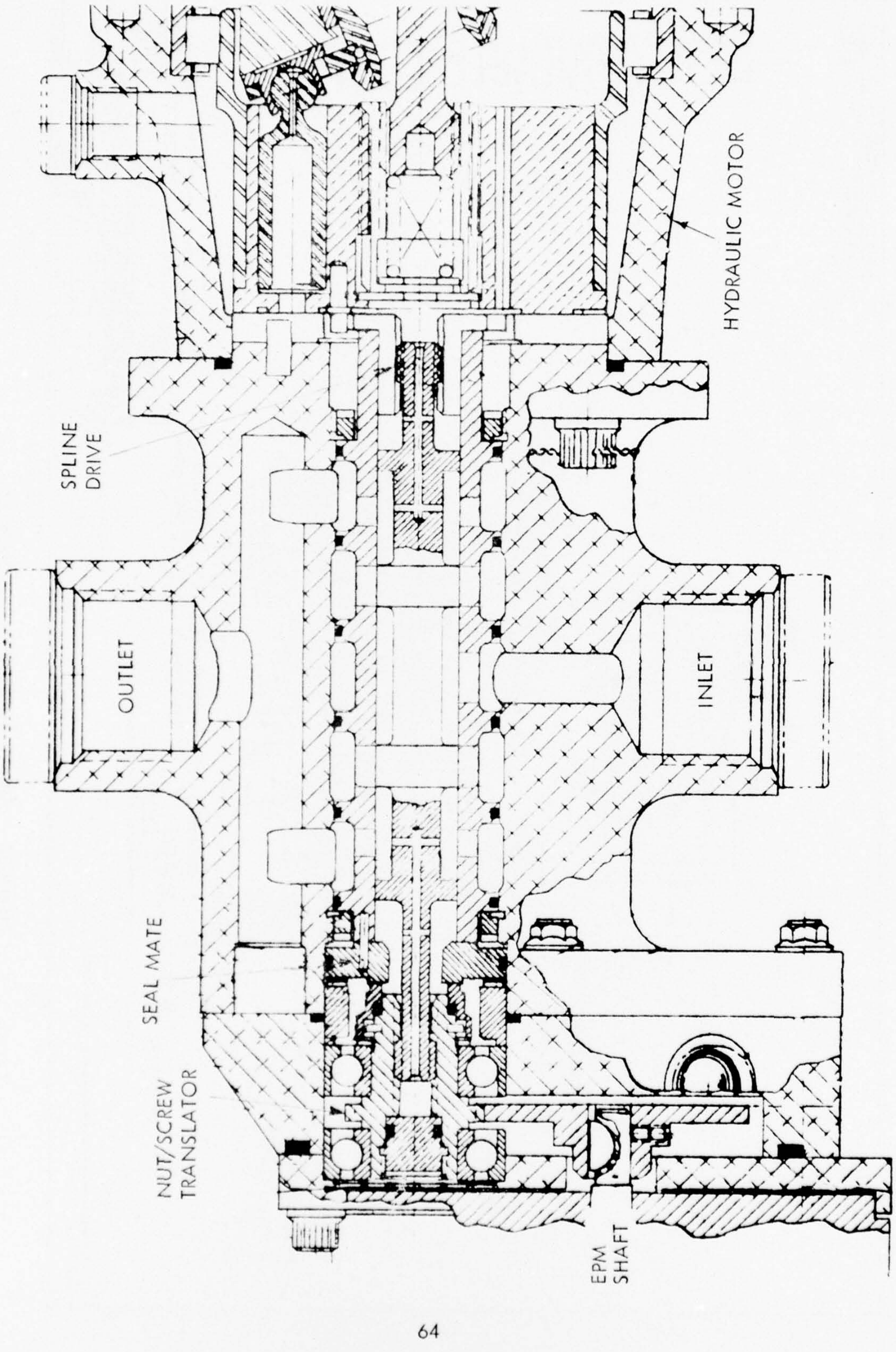


Figure 34 Abex Drawing of EHPM (Sheet 3)

SECTION VI
PROTOTYPE EHPM TESTING

Component Tests

Input Drive and Pulse Motor - The input drive system and EPM were developed and tested as a single component. The purpose of these tests was to develop and demonstrate the speed, ramping, and position control requirements of the prototype EHPM actuation system.

The block diagram of the Input Control System is shown in Figure 35. The breadboard test arrangement used as a first verification of software is shown in Figures 36 and 37. A handle geared to the input encoder simulated the pilot's flap control.

Voltage signals simulating the pressure transducer were fed into the interface board to verify that software caused a reduction in EPM speed when pressure fell below 1500 psi, stopped the EPM when below 1200 psi, and provided adequate delay before restarting so that unstable cycling did not occur. This test arrangement served as a test bed for developing and verifying the software and control logic shown in Appendix B.

The apparatus proved to be a most useful tool, and it is believed that the behavior of the EPM output, which is readily observed, can be easily related to the behavior of the EHPM installed in the system, assuming that the hydraulic motor follows the EPM.

Using the control logic and the software program described in Appendix B, the input breadboard arrangement was operated and the behavior of the EPM output was visually observed for proper response. Signals to represent the pressure transducer were fed in using a pulse generator and observed via an oscilloscope. Behavior of the EPM due to simulated pressure signals was visually observed at the EPM output.

For the breadboard development tests there was no data recorded. Success or failure of the test was judged by the observed behavior of the EPM output based on the motion of the input handle and the simulated pressure signals. Failures indicated that changes in the program were required and these were made as the need developed so that when the system behaved properly the testing was considered complete and successful to the extent achieveable using the breadboard arrangement.

Speed Control - The speed control requirements were demonstrated. The crew control was moved to some intermediate flap position, the stepper motor accelerated from 0 to 10,000 PPS (2,000 RPM) and ran at constant speed until the selected position was reached, then decelerated and stopped. The control was returned to flaps up position and the stepper motor reversed direction and ran to the UP position with controlled acceleration and deceleration ramps. Effective ramp control was demonstrated when reversing direction of rotation.

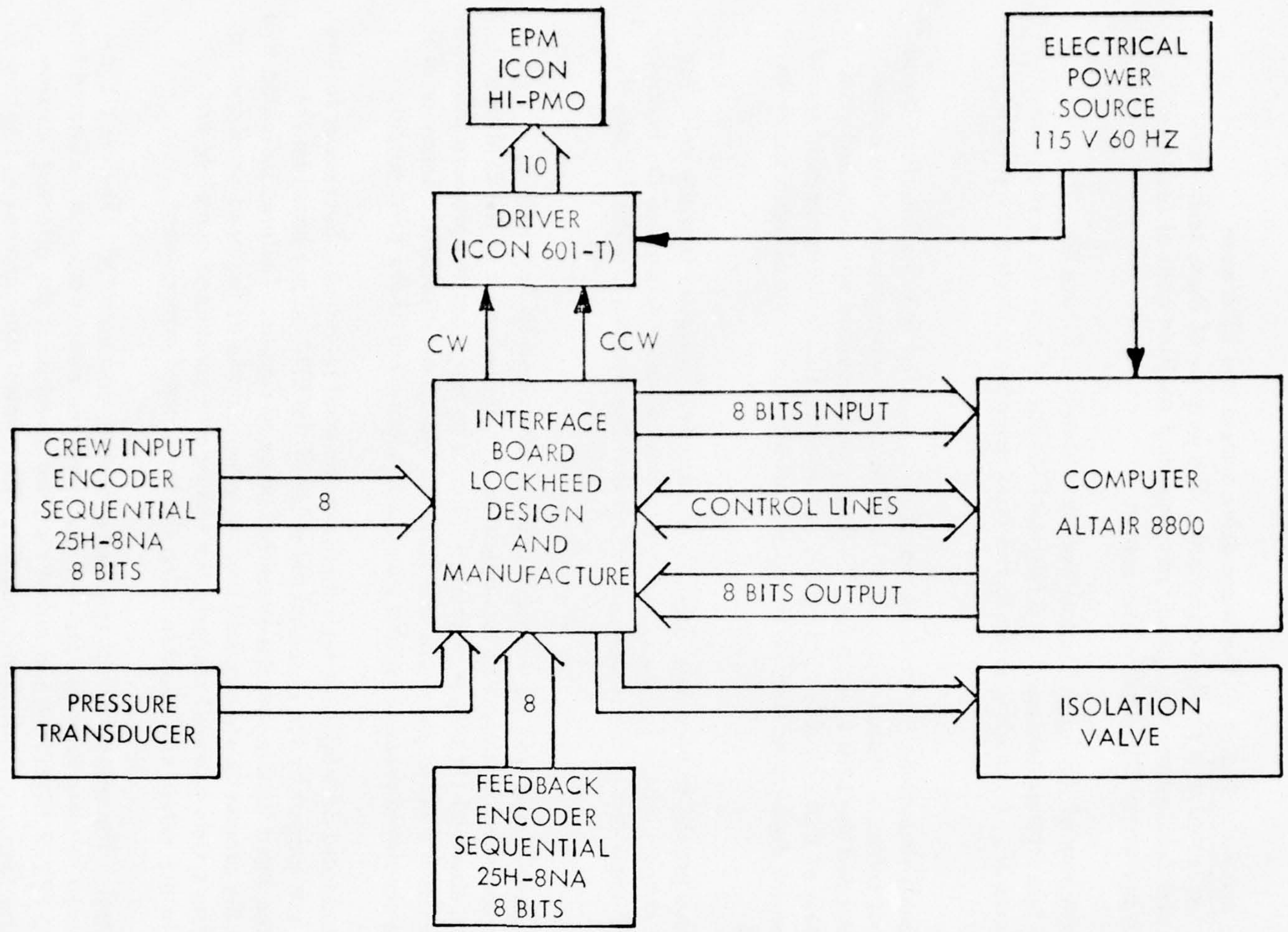


Figure 35 Input Control System

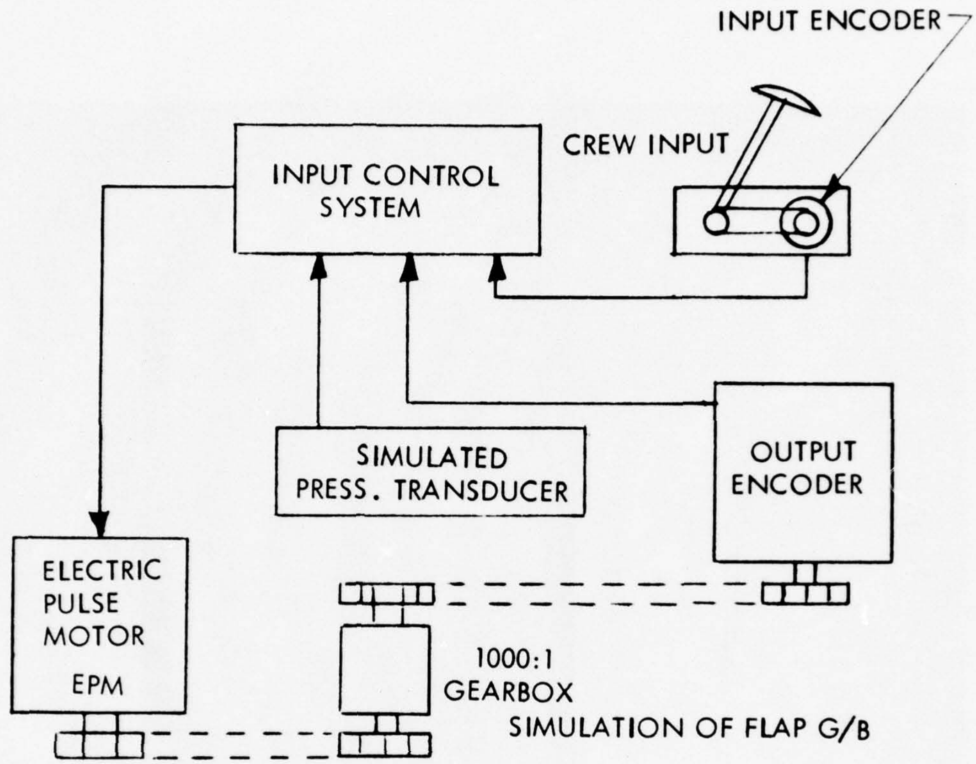


Figure 36 Test Arrangement of Input Control System

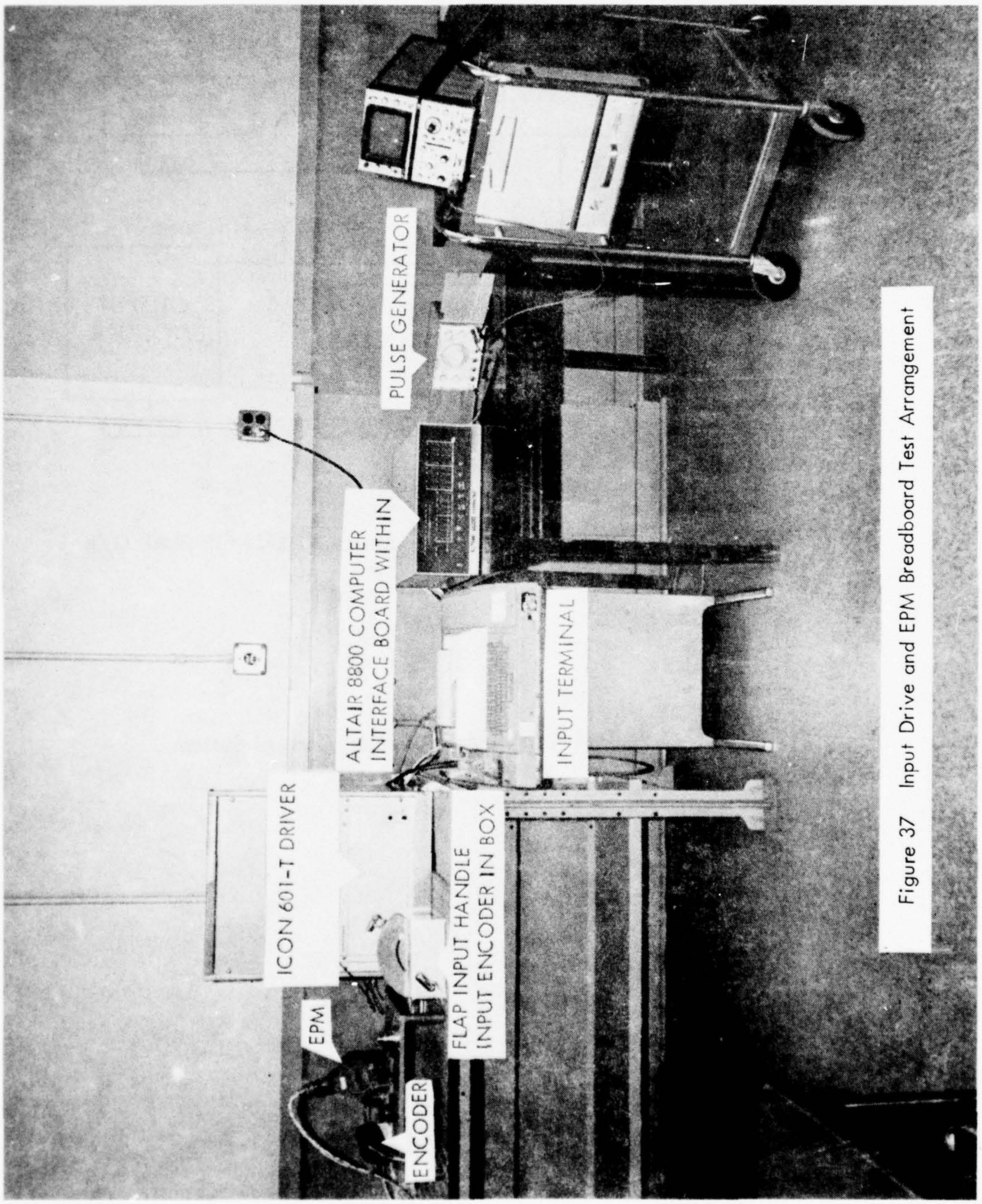


Figure 37 Input Drive and EPM Breadboard Test Arrangement

A variable voltage signal simulating a pressure transducer was delivered to the interface board of the input control system. At a level above 1500 PSI, the stepper motor output speed was 10,000 PPS or 2,000 RPM. As the level was decreased below 1500 PSI, the following occurred as a function of inlet pressure.

- Stepper motor speed decreased to 1000 PPS at a pressure level less than 1500 PSI.
- As the pressure level continued to decrease, the stepper motor stopped for pressure less than 1200 PSI.
- At a pressure level of 1200 PSI and below, a 28-volt signal to operate the system shutoff valve was delivered.
- On increasing pressure, the shutoff valve signal was removed at 1500 PSI. The stepper motor ramped back to limit speed (10,000 PPS). A variable time delay between system off and back on because of pressure control was demonstrated.

Position Control - The crew input control was moved to any position and the output moved to a corresponding position. The positions of takeoff and approach (737 revolutions of EPM)and landing (921 revolutions of EPM) were demonstrated. Various positions were selected in both extending and retracting flap modes. Reversing while in motion was accomplished.

During the EHPM assembly tests, which are described later, it was suspected that the EPM was not performing up to specification. A "pull-out" test was devised to measure the torque required to stop the EPM, i.e., to pull it out of synchronous speed (it stops). This torque was measured using the apparatus shown in Figure 38. The EPM was first accelerated to the desired speed. The torque was increased by applying by hand a small force to the tail string. The number of turns of string around the wheel was made sufficient so that the control force on the tail string required to achieve the larger force to the spring scale was insignificant compared to the large force. The force was gradually increased while observing the spring scale. The scale reading when the EPM suddenly stopped was recorded as the "pull-out" torque. These tests revealed that the EPM torque when driven by the Icon 601-T driver was considerably below specification. Analysis revealed the need for a different power supply with a current limiting feature which would vary voltage to the EPM as necessary to keep the current high as pulse rate increased. Using a power supply (Lambda Electronics Corp., Model LE 104 FM) which incorporates this feature and with a peak voltage capability of 36 volts the performance of the EPM was significantly improved as shown in Figure 39.

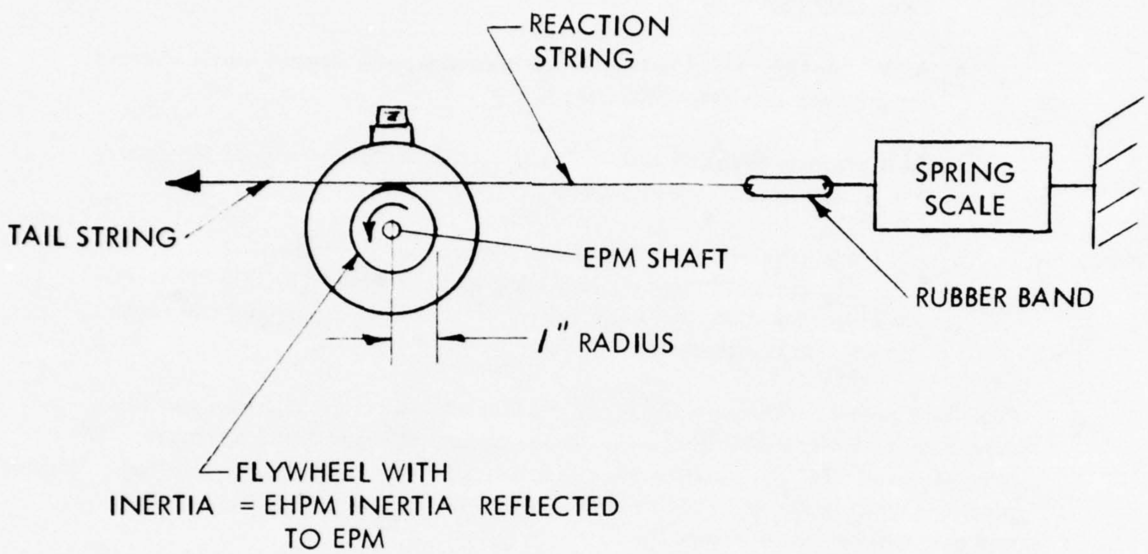


Figure 38 Test Arrangement for Measuring "Pull Out" Torque of EPM

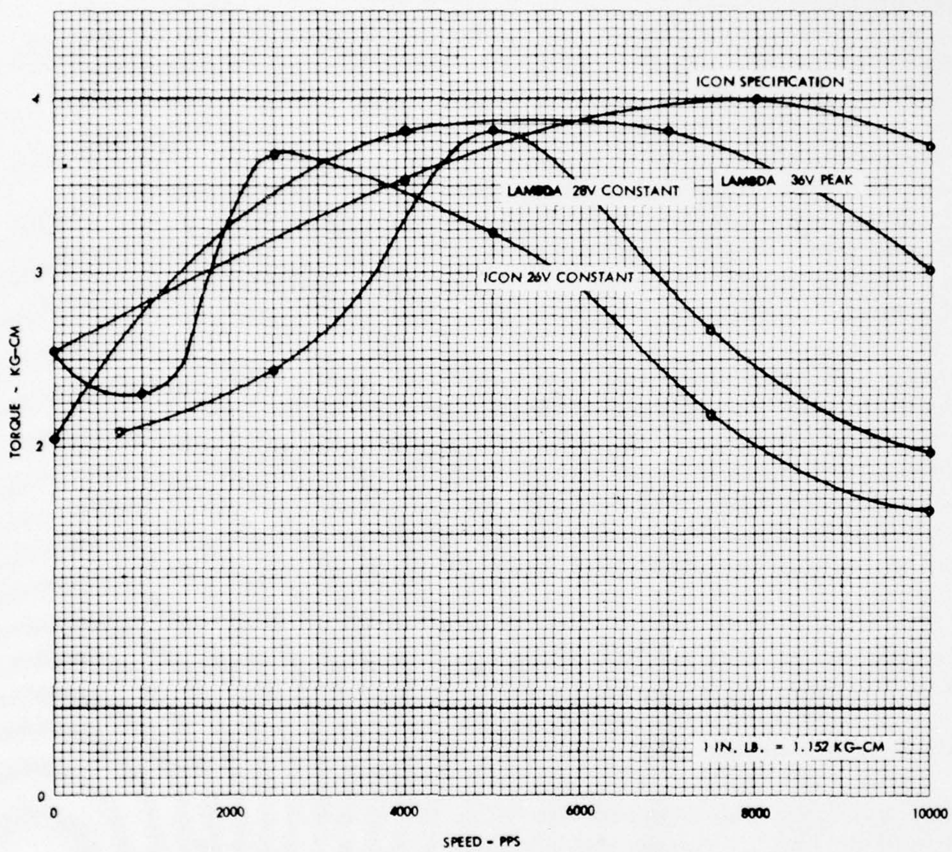


Figure 39 EPM Pull Out Torque

Valve - Hydraulic Motor Tests - These tests consisted of conventional hydraulic motor tests using a conventional port cap, flow tests of the valve (performed at ABEX) and drag torque tests using a DC motor in lieu of the EPM to drive the valve. Two significant problems occurred during these tests. If the valve reaches its maximum travel and bottoms, the nut thread jams and cannot be broken loose by the electric motor. The motor torque plus inertia creates a greater torque for jamming the thread than the motor torque can provide for unjamming the thread. Also, on two occasions the valve spool seized in the sleeve. Chrome plating the spool and opening the clearance between the spool and sleeve appeared to solve this problem. The thread jamming problem is solvable by conventional means, i.e., providing stops that apply a tangential load between the two threaded parts instead of an axial load. Designs to accomplish this were investigated and found to be reasonable. However, no fix was installed in the test unit. In a production unit non-jamming threads will be required.

The drag torque tests revealed an unexpected directional quality. This asymmetry is caused by the pressure load on the EPM end of the valve. As the valve moves away from the EPM the return flow causes higher pressure on the left spool-end than on the right spool-end which is vented through the motor case. The pressure force on the spool is opposite to the flow forces and causes less load to be reflected to the EPM. The phenomena is beneficial, and is a desirable design objective for the other direction. The results of this test are discussed under Durability Tests where the test was repeated.

Drag torque measurements were made using the test arrangement of Figures 40 and 41. Voltage to the DC motor was increased until the desired output speed of the hydraulic motor was reached and then current, speed, and voltage were recorded. The torque output of the DC motor was available from the DC motor calibration tests.

The DC motor calibration tests were performed using the apparatus described previously for EPM "pull-out" tests. Voltage was increased in steps and then torque was increased until the motor slowed to a selected speed. Voltage, speed, torque, and current were recorded. Values were selected to cover the full range of loads and speeds which encompass the specification capability of the EPM.

Flow grinding of the valve provided .0002 to .0004 inch overlap and a neutral cylinder pressure of 250 to 500 psi with 3000 psi supply. The flow gain characteristic is shown in Figure 42. Above .080 inch spool displacement the test was conducted at decreasing supply pressures to maintain a constant flow. The pressure required to produce 27 gpm flow at .010 increments between .080 and .156 was recorded. This data is shown in Figure 43.

EHPM Assembly Tests - The EHPM assembly is shown in Figure 44. Using the test arrangement shown in Figures 45 and 46, the assembly was tested as follows:

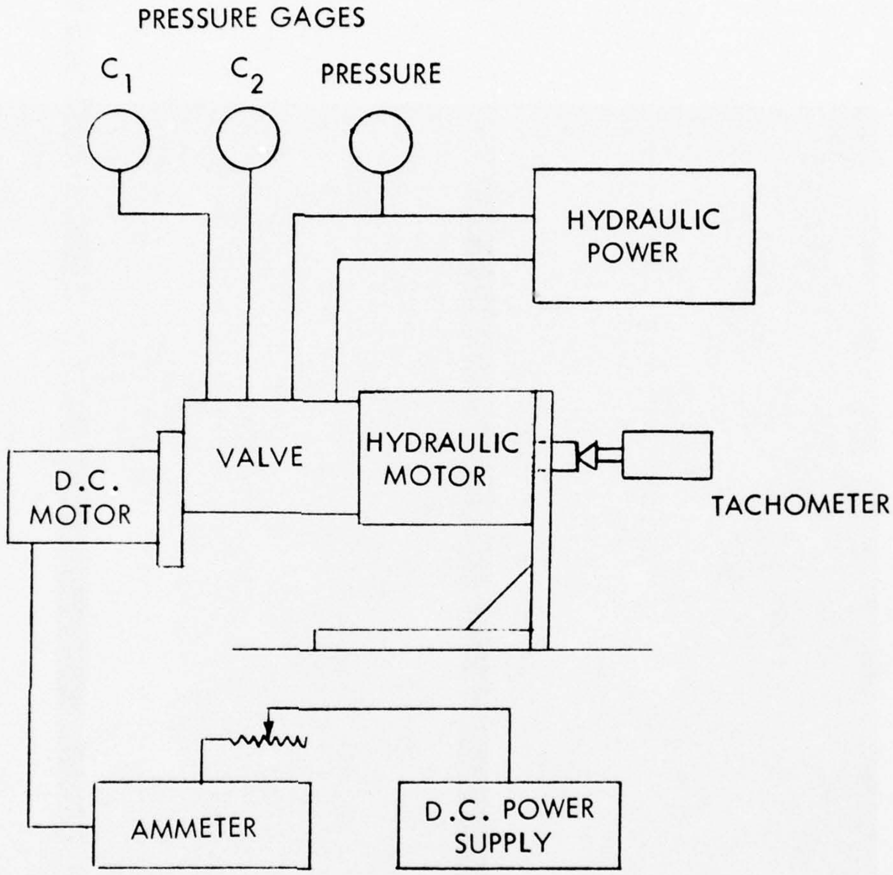


Figure 40 Test Arrangement for Valve Drag Torque Measurements

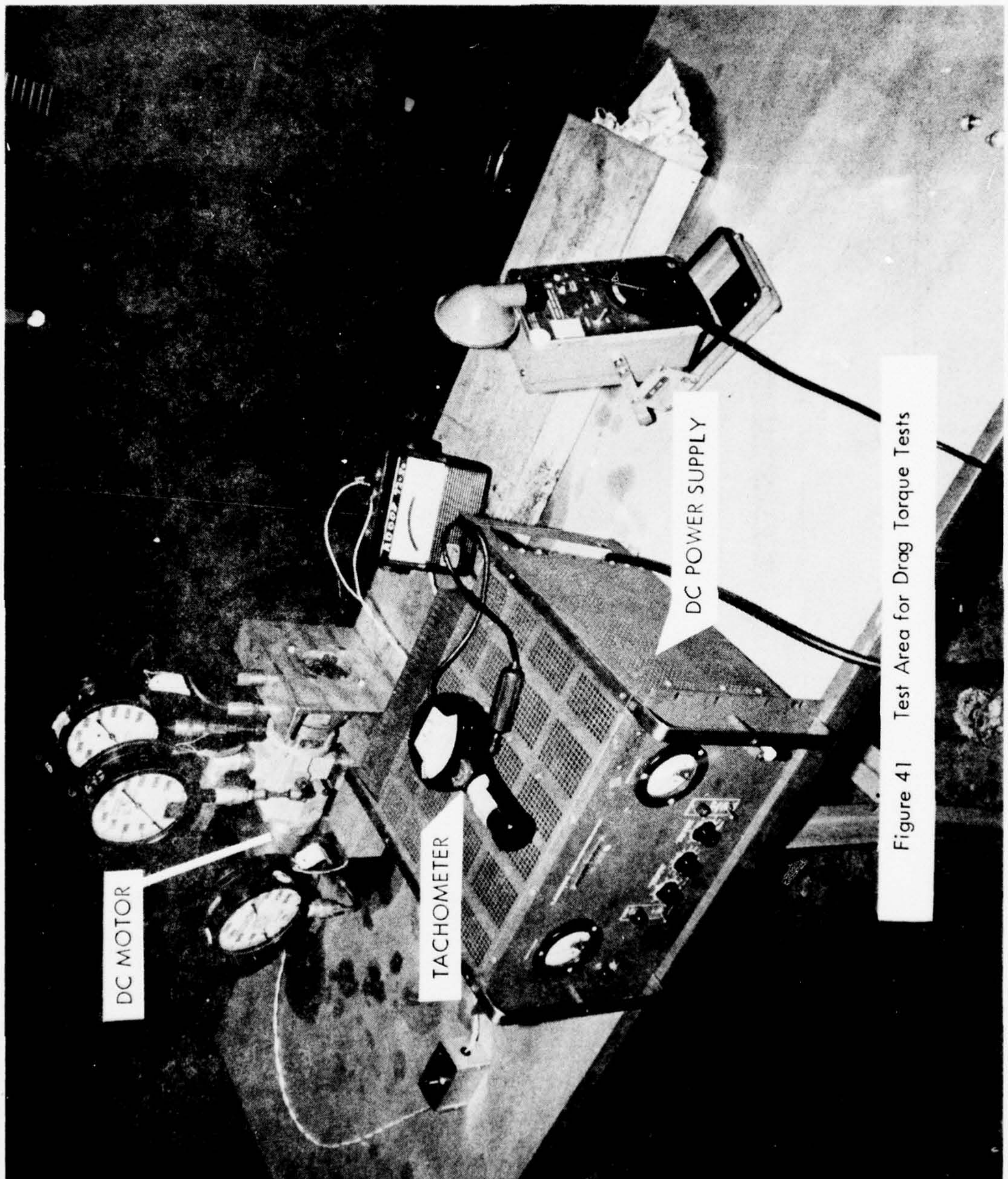


Figure 41 Test Area for Drag Torque Tests

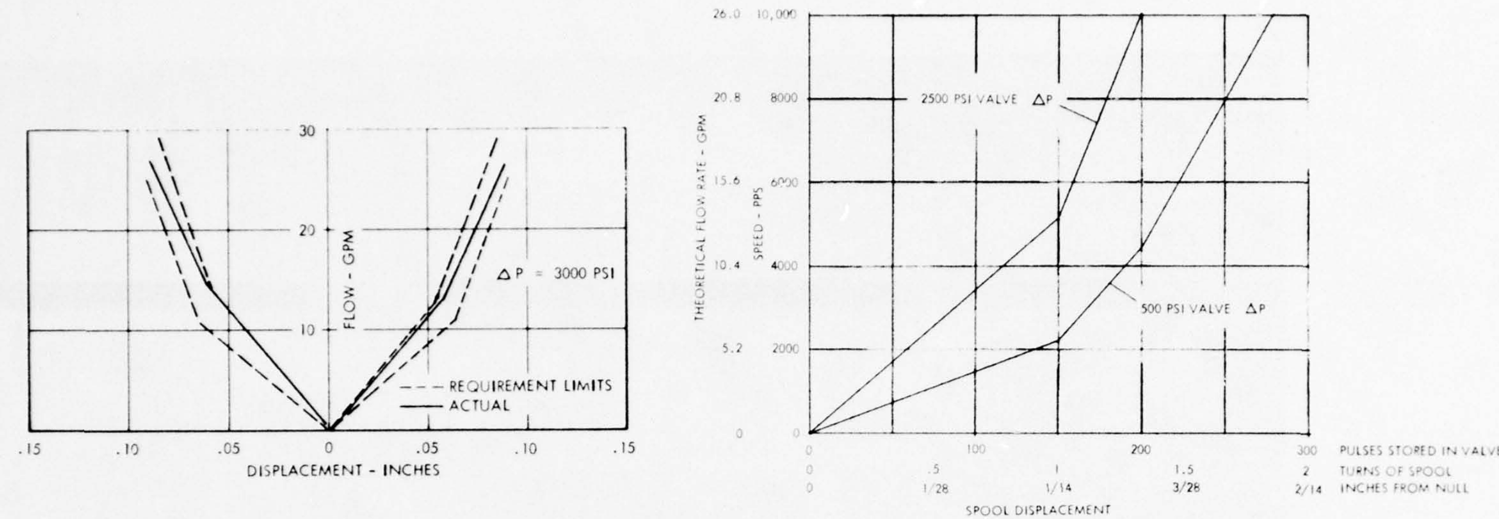


Figure 42 Valve Flow Gain Characteristics

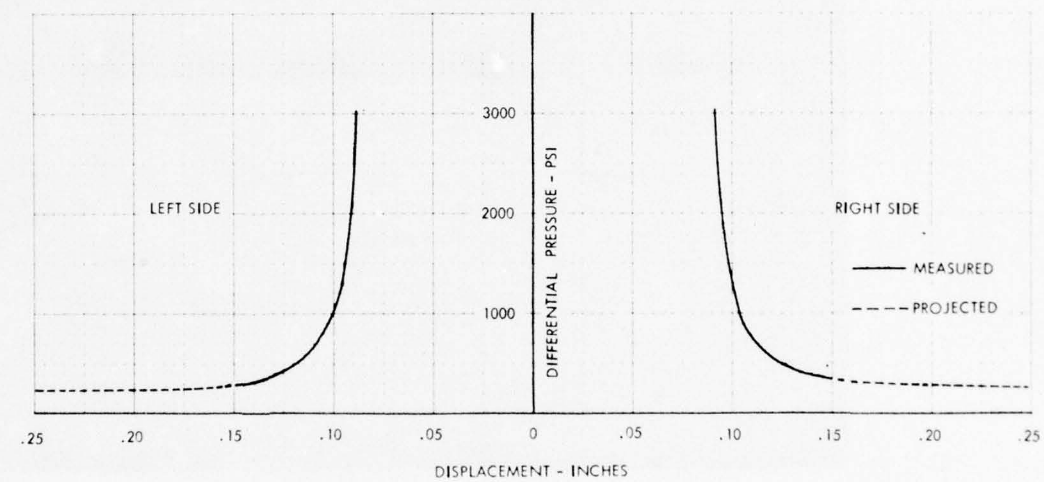


Figure 43 Valve Pressure Drop with 27 GPM

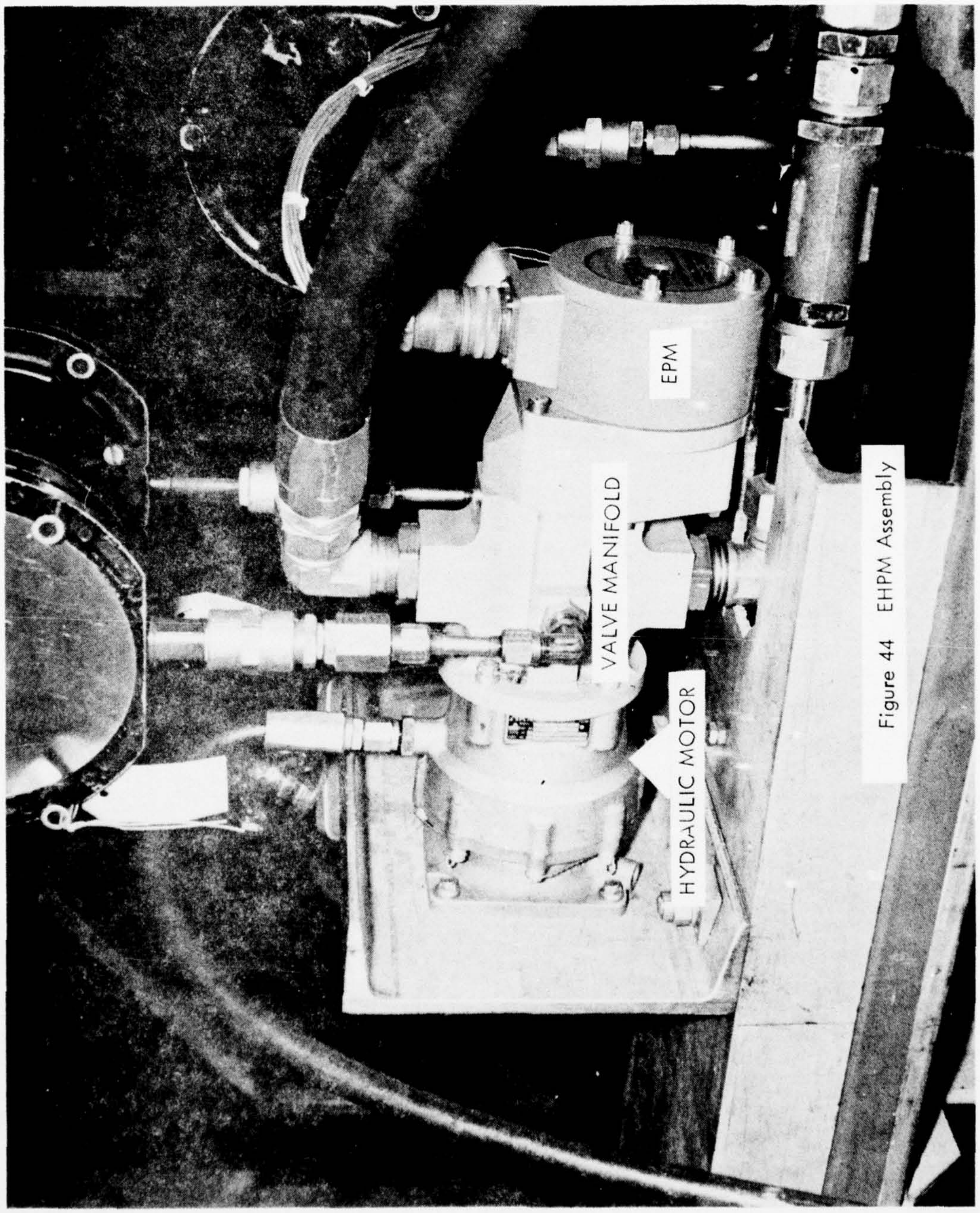


Figure 44 EHPM Assembly

1. FILTER
2. MANUALLY ADJUSTABLE RESTRICTOR
3. CHECK VALVE
4. PRESSURE GAGE 0-5,000 PSI
5. HYDRAULIC LOAD MOTOR
6. EHPM HYDRAULIC MOTOR
7. EHPM VALVE
8. EPM
9. COOLER
10. PRESSURE GAGE 0-600 PSI

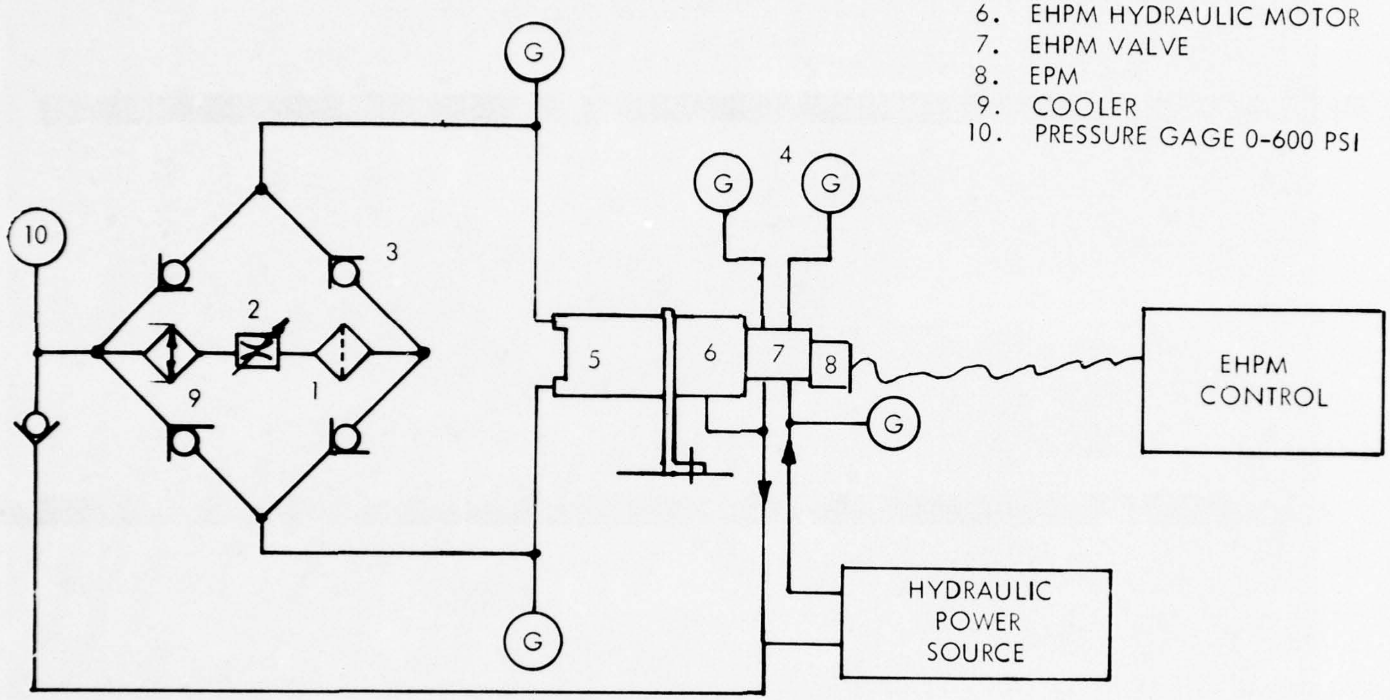


Figure 45 Functional Performance Test Arrangement

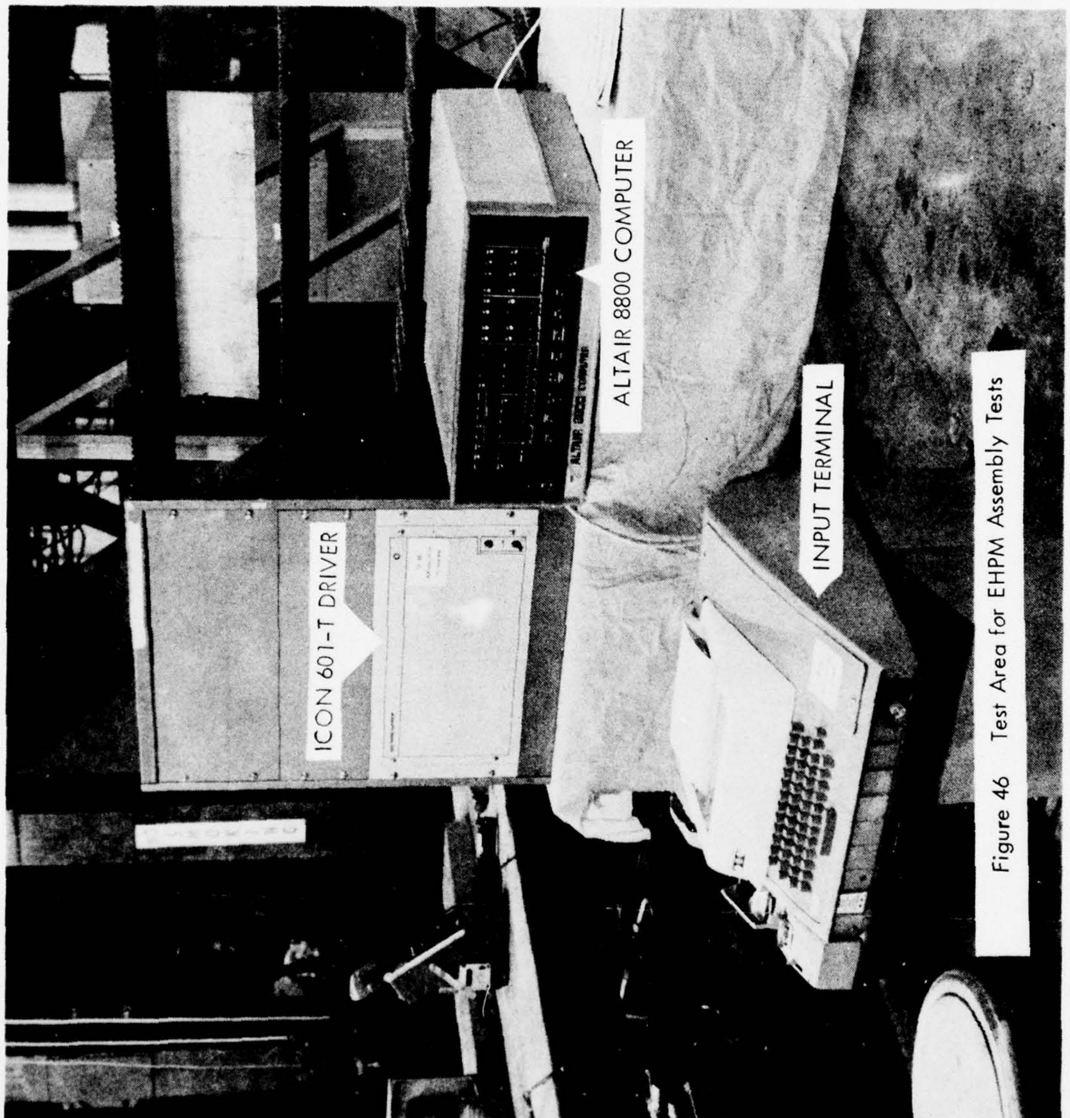


Figure 46 Test Area for EHPM Assembly Tests

No Load - With the load motor (item 5 of Figure 45) and its circuit removed so that the output shaft of the hydraulic motor (item 6) was exposed, attempts were made to accelerate the EHPM to speed. A hand-held tachometer was used on the hydraulic motor output shaft to measure speed.

These tests were attempted initially with an acceleration ramp which consisted of 40 speed values equally divided between 0 and 10,000 pps, i.e., equal steps of 250 pps. The time increment between speed changes was established as a variable to be readily and easily alterable. This arrangement is referred to as a linear ramp. The following results were obtained:

Time Increment, ms	Speed Reached, rpm	Time to Speed, sec
20	2000	0.40 (.02 x 20)
32	2400	0.77 (.032 x 24)
48	2500	1.20 (.048 x 25)
96	3000	2.88 (.096 x 30)
192	3500	6.72 (.192 x 35)

The ramp was revised to provide 83 values so that the speed differences between adjacent values in the acceleration table could be smaller as the speed increased, i.e., a nonlinear ramp (simular to an exponential ramp).

The ramp used is illustrated in Figure 47 and is compared with two exponential curves and a sine wave which approximate the selected ramp at the upper end. The same ramp was used for deceleration.

In accelerating to a limit speed the control established an initial pulse frequency (R) of 750 pps, delayed an increment of time, Δt, established R = 1000 pps, delayed Δt, established R = 1200, delayed Δt, etc. until the limit speed was reached. The limit speed was a settable value in the program. The values 750, 1000, 1200 -- 10,000 were in effect stored in memory as a table. To accelerate, the table was ripped through in increasing value and to decelerate was ripped through in decreasing value. The delay increment, Δt, was also a settable value in the program. After being set it was the same for every speed value. It is pointed out that for limit speeds which were less than 10,000 pps, the limit speed was not approached gradually. For example, if the limit speed were 6000 pps, 30 time increments were used to reach limit speed along the curve of Figure 47 and then the frequency remained constant at 6000 pps.

With the revised ramp the unit would still not reach a 4000 rpm limit speed. Limit speed, a setting in the program, was reduced in steps until the unit could achieve limit speed. After achieving this speed it would sometimes stop, indicating that acceleration was not the problem and that EPM torque was marginal. A test was then devised to measure EPM "pull-out" torque. This test was discussed earlier under the input system. Low EPM torque was found to be the problem. A revision to the DC power supply was made to correct the problem. With the revised power supply (Lambda)

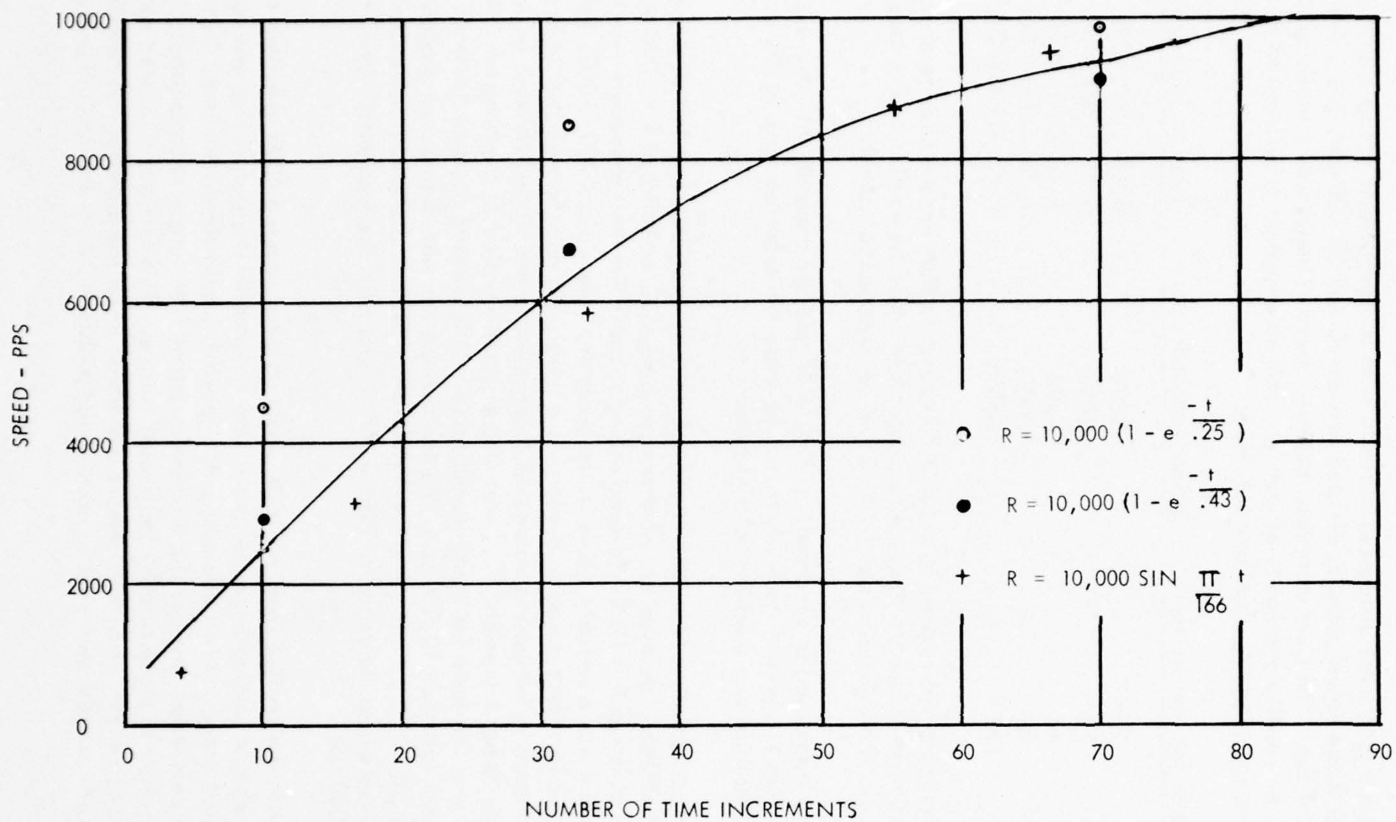


Figure 47 Non-Linear Acceleration Ramp Compared with Formalized Curves

4000 rpm could be achieved with 30 volts in the CW direction and 3200 rpm with 35 volts in the CCW direction. It would reach these speeds with a time increment as low as .007 sec (time to reach 4000 rpm = 83 increments x .007 sec. = .58 sec.).

Load - With the load motor installed (as in Figure 45) the EHPM was accelerated to a selected speed and the load was applied using a manually operated needle valve (item 2 of Figure 45). Then without changing the position of the load valve the EHPM was stopped and repeatedly started and stopped in both directions. A load of 2350 psi differential between C1 and C2 was the maximum used. Data were recorded from gages for some of these tests and with a brush recorder for others. Deceleration ramps were not used for these tests.

The unit performed slightly better under load, achieving 4000 rpm in the CW direction and 3840 rpm in the CCW direction with a time increment to .004 sec. consistently and to .003 sec. not consistently.

Leakage Tests - With a fluid temperature of 100°F, C_1 and C_2 ports open and 3000 psi applied to the pressure port and with leakage from C_1 and C_2 visually equalized by manually moving the hydraulic motor output shaft the combined leakage from C_1 and C_2 was 890 cc/min.

With C_1 and C_2 closed and their pressures equalized by manually moving the hydraulic motor output shaft, the combined leakage from the return and motor case was 900 cc/min. (0.24 gals/min).

Iron Bird Tests

The EHPM control circuit illustrated in Figure 48, controlled by the digital computer, was used to operate the C-5 Iron Bird flap system, Figures 49 through 55, with no load, intermediate load, and maximum load. Initially the return from the EHPM was routed into the normal system return which is exposed to boost pump pressure that is nominally 100 psi. The unit would not operate satisfactorily due to return pressure loads on the nut bearings which must be rotated by the EPM. When the return was routed directly to the reservoir the unit could achieve 100% speed (4000 RPM) in the extend mode and 92% speed in the retract mode. The unit started and stopped smoothly at all positions of the flap and with the various loads. The acceleration and deceleration ramps could be readily altered and the effect on system performance promptly evaluated from the data recordings. The speed reduction feature, which causes speed to reduce when pressure falls below set point (1500 psi) and to stop and apply brakes when below brake set point (1200 psi) wait 2 seconds and then restart, worked well, except that operation at the 1500 psi set point caused a pressure (±125 psi) and flow (±1.5 gpm) pulsation of approximately 1 cps while the average flow and speed was decreased to maintain the 1500 psi inlet pressure. The motion of the flap mechanism was not severe and was controlled by the program. The EHPM responded as programmed by stepping up in speed and then down in speed to maintain the pressure setting. Although this mode of operation is not normal it does occur when system flow demand is greater than system flow capacity. It is believed that the pulsation can be removed by programming, but time did not permit trial of the program changes.

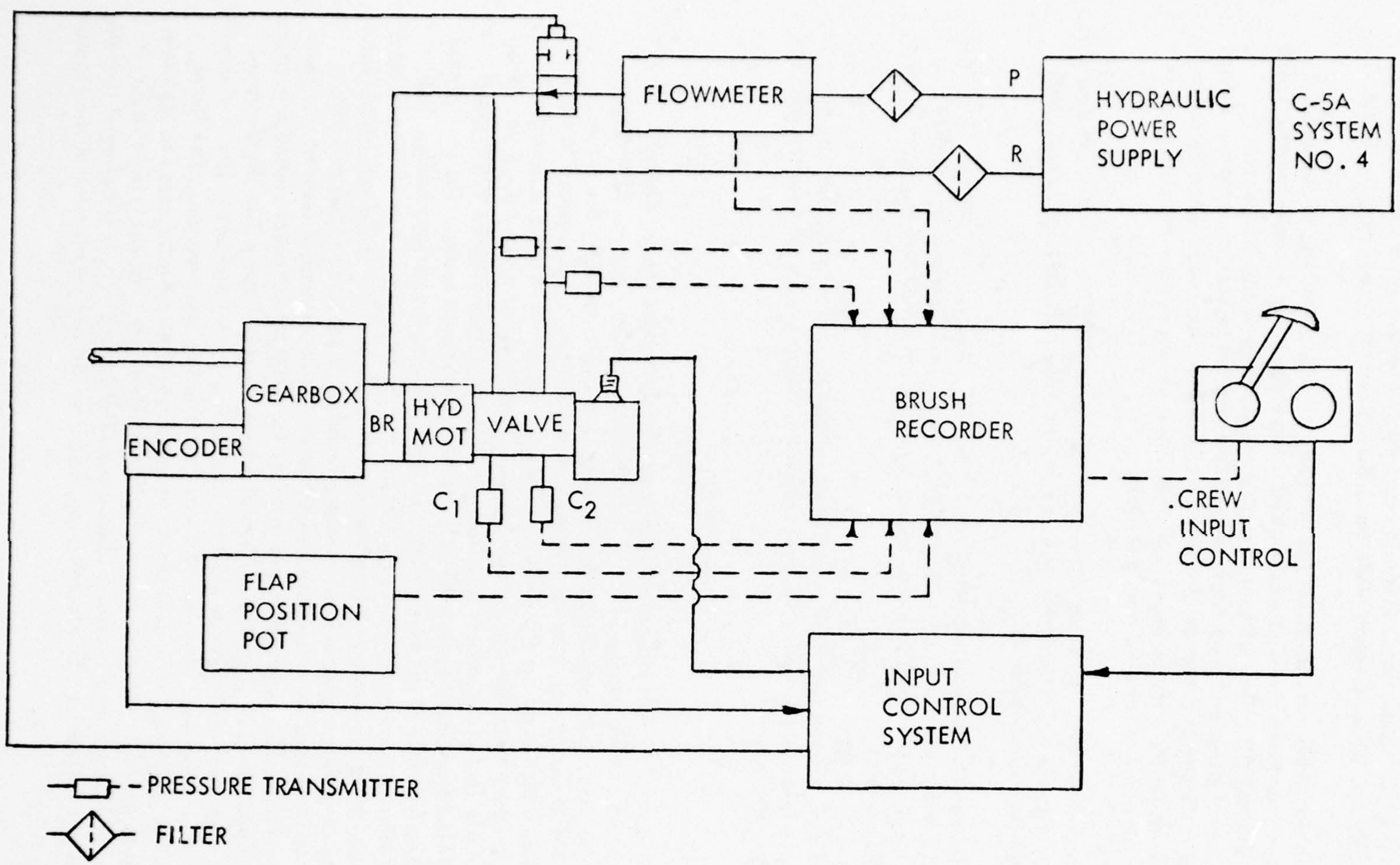


Figure 48 Iron Bird Tests Instrumentation Arrangement

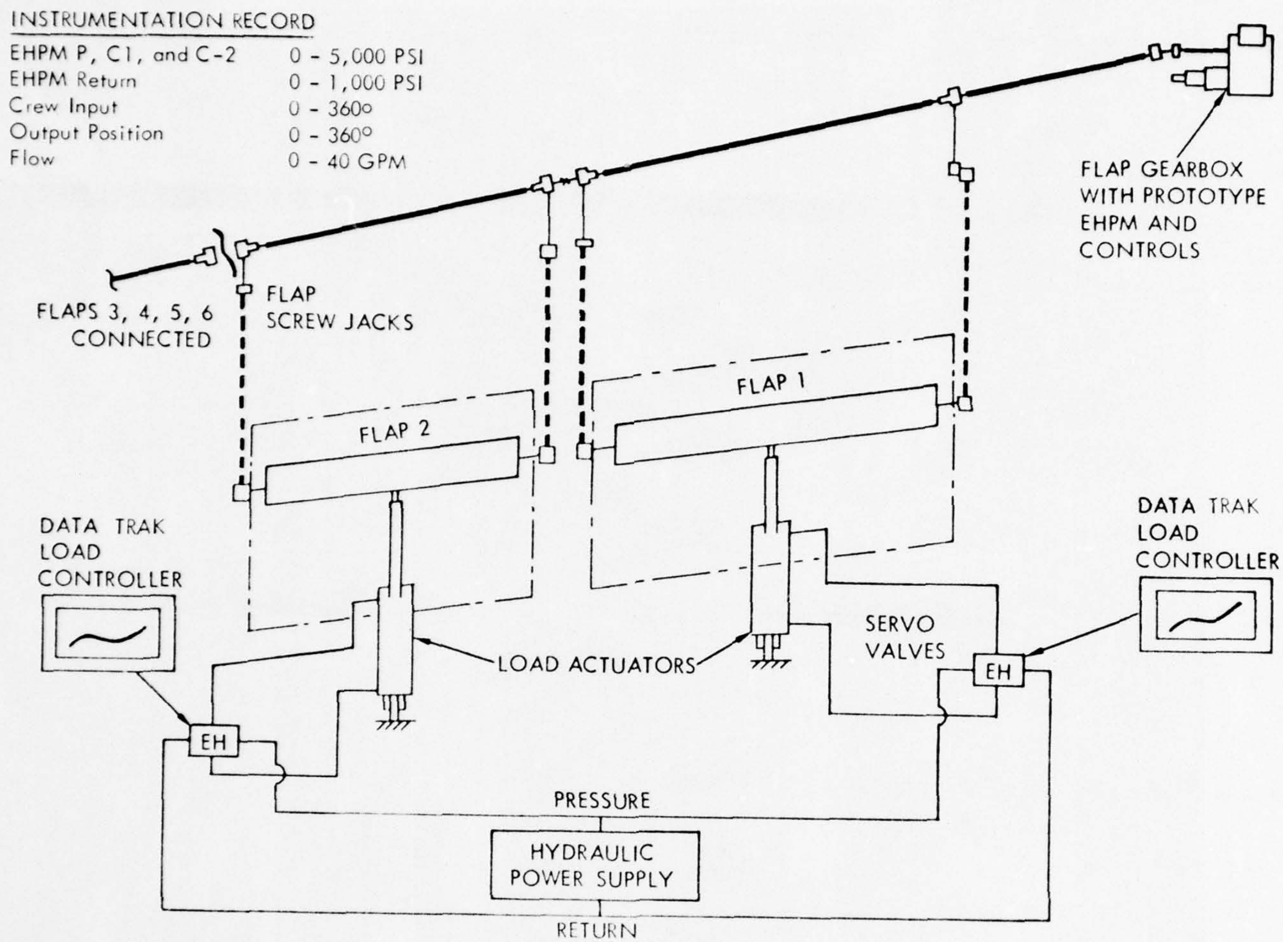


Figure 49 Iron Bird Test Arrangement - Flap System

Figure 50 C-5 Iron Bird Flap and Loading Mechanism

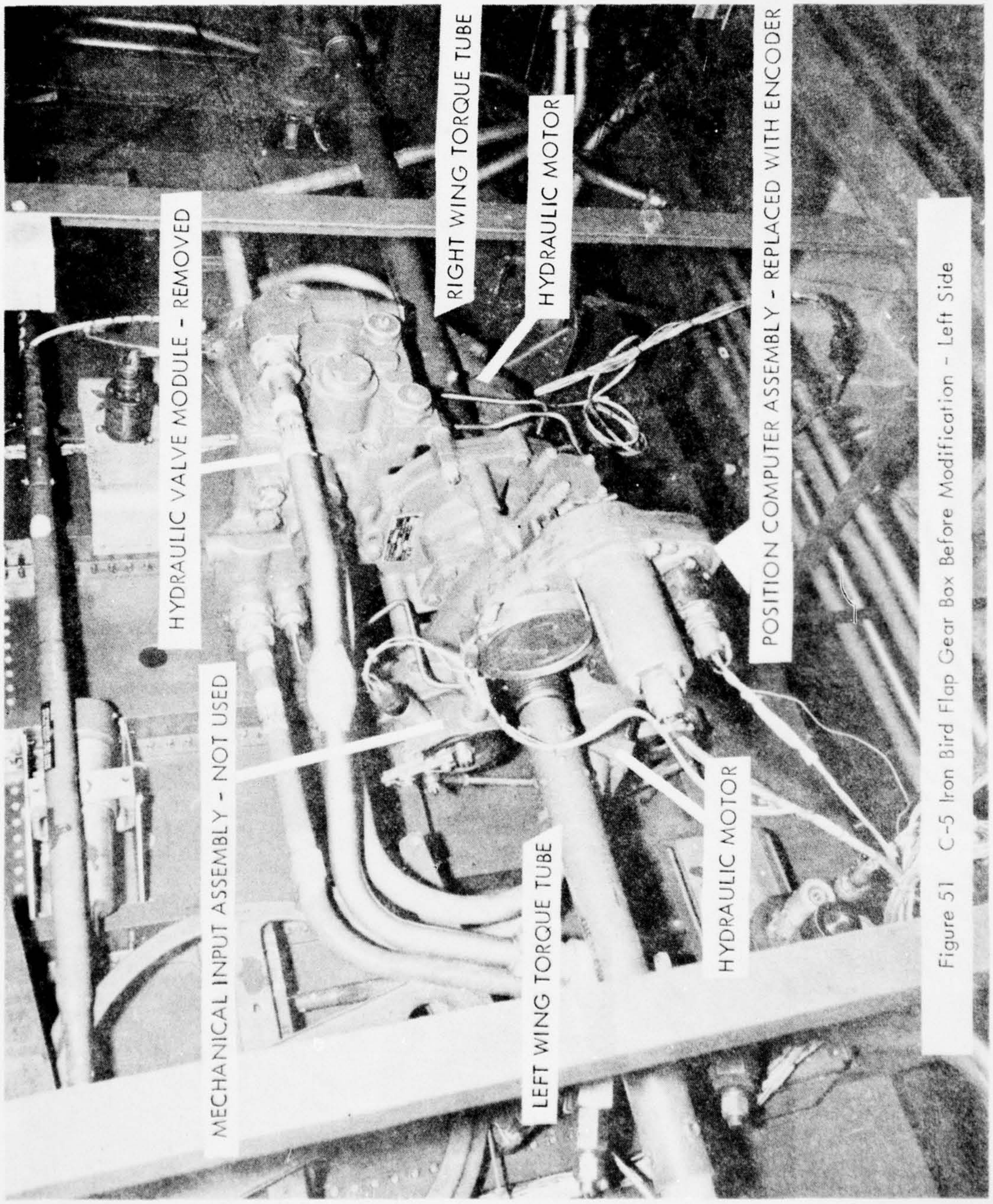


Figure 51 C-5 Iron Bird Flap Gear Box Before Modification - Left Side

AD-A042 377 LOCKHEED-GEORGIA CO MARIETTA F/G 1/3
INVESTIGATION OF ELECTRO-HYDRAULIC PULSE MOTORS FOR AIRCRAFT UT--ETC(U)
MAY 77 E W RUMRILL, F D LEWIS F33615-75-C-2005
UNCLASSIFIED LG77ER0099 AFAPL-TR-77-14 NL

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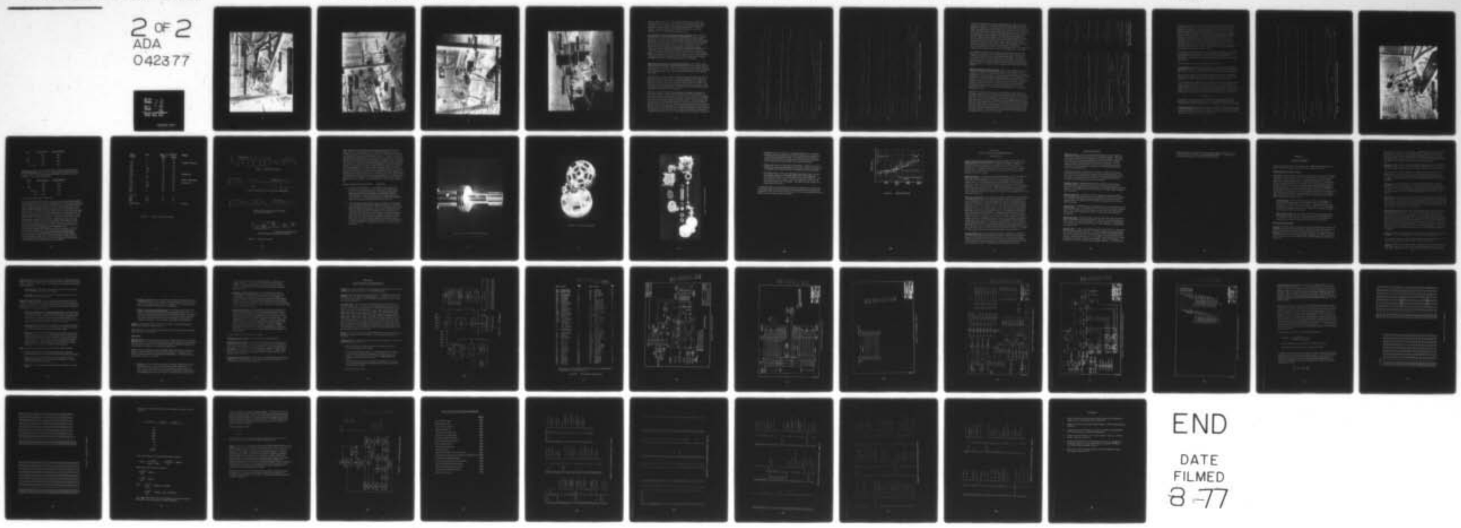

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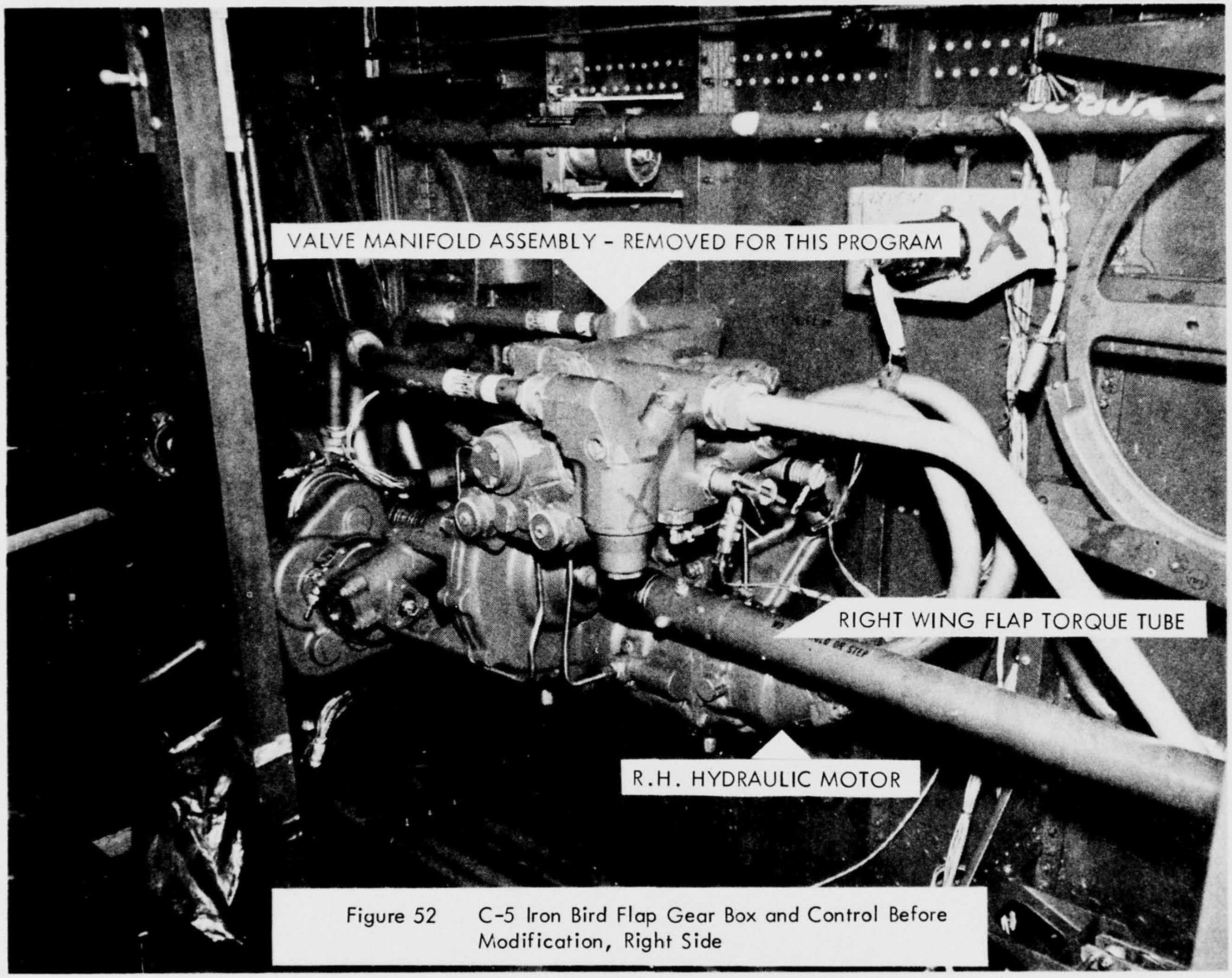


Figure 52 C-5 Iron Bird Flap Gear Box and Control Before Modification, Right Side

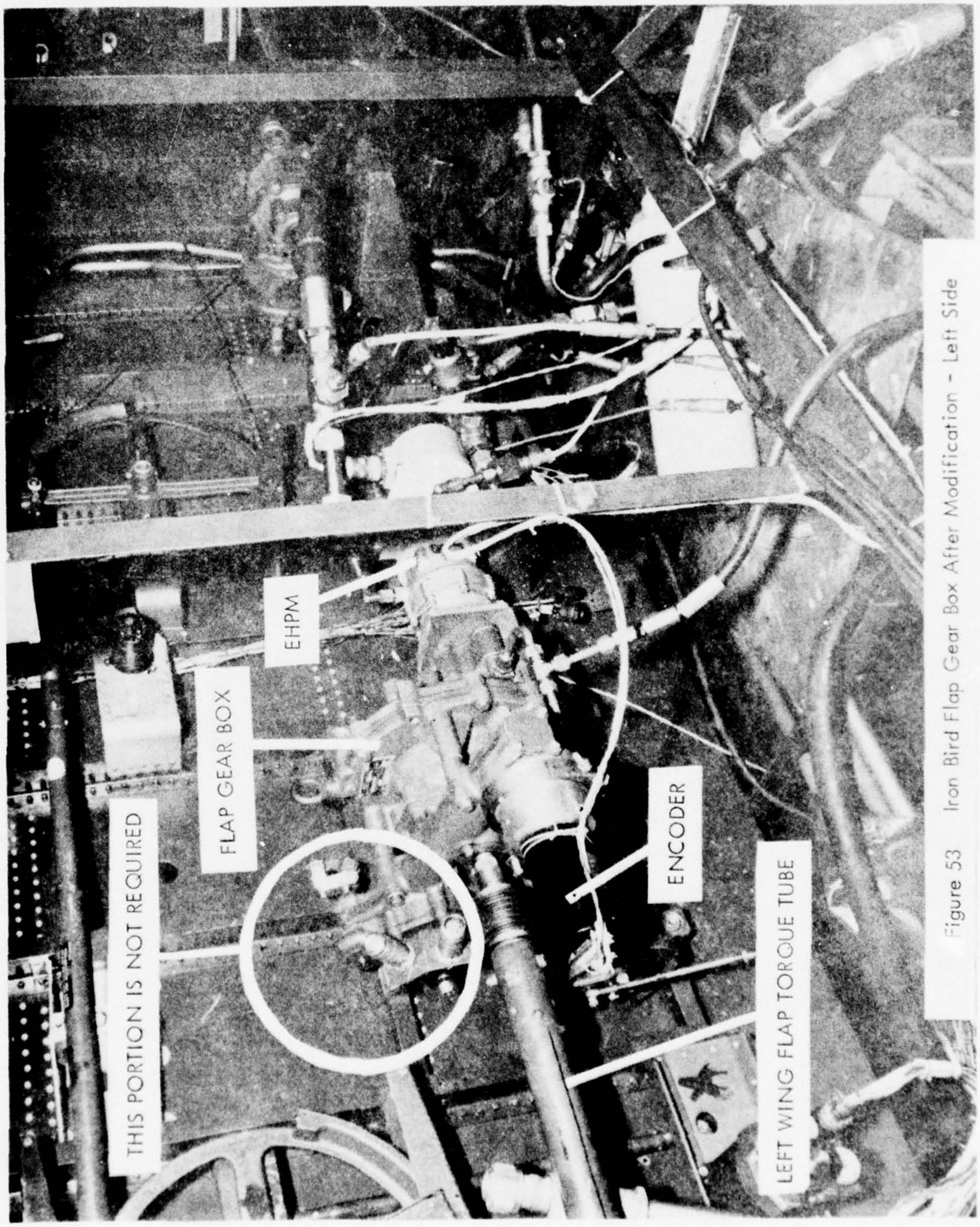


Figure 53 Iron Bird Flap Gear Box After Modification - Left Side

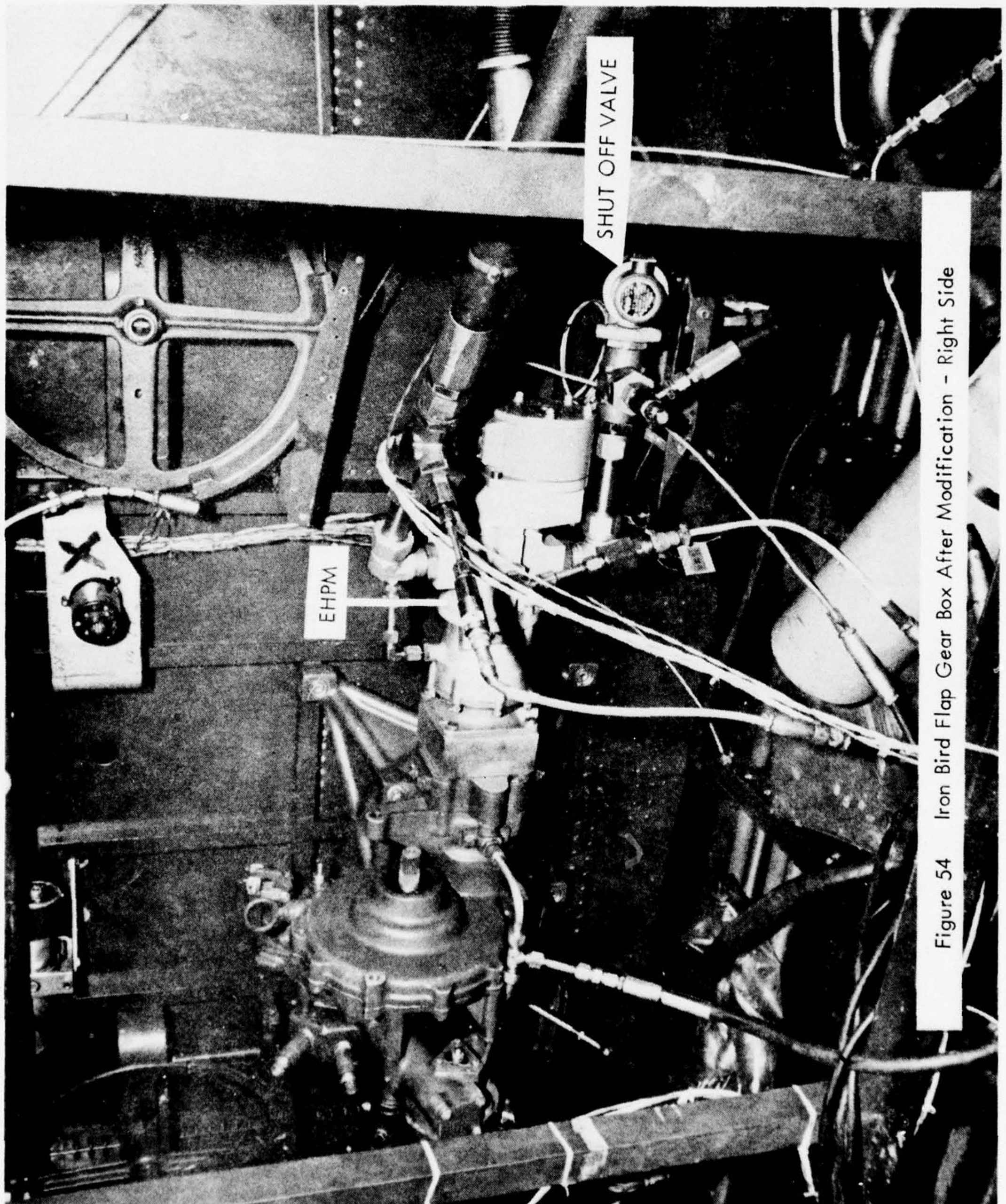


Figure 54 Iron Bird Flap Gear Box After Modification - Right Side

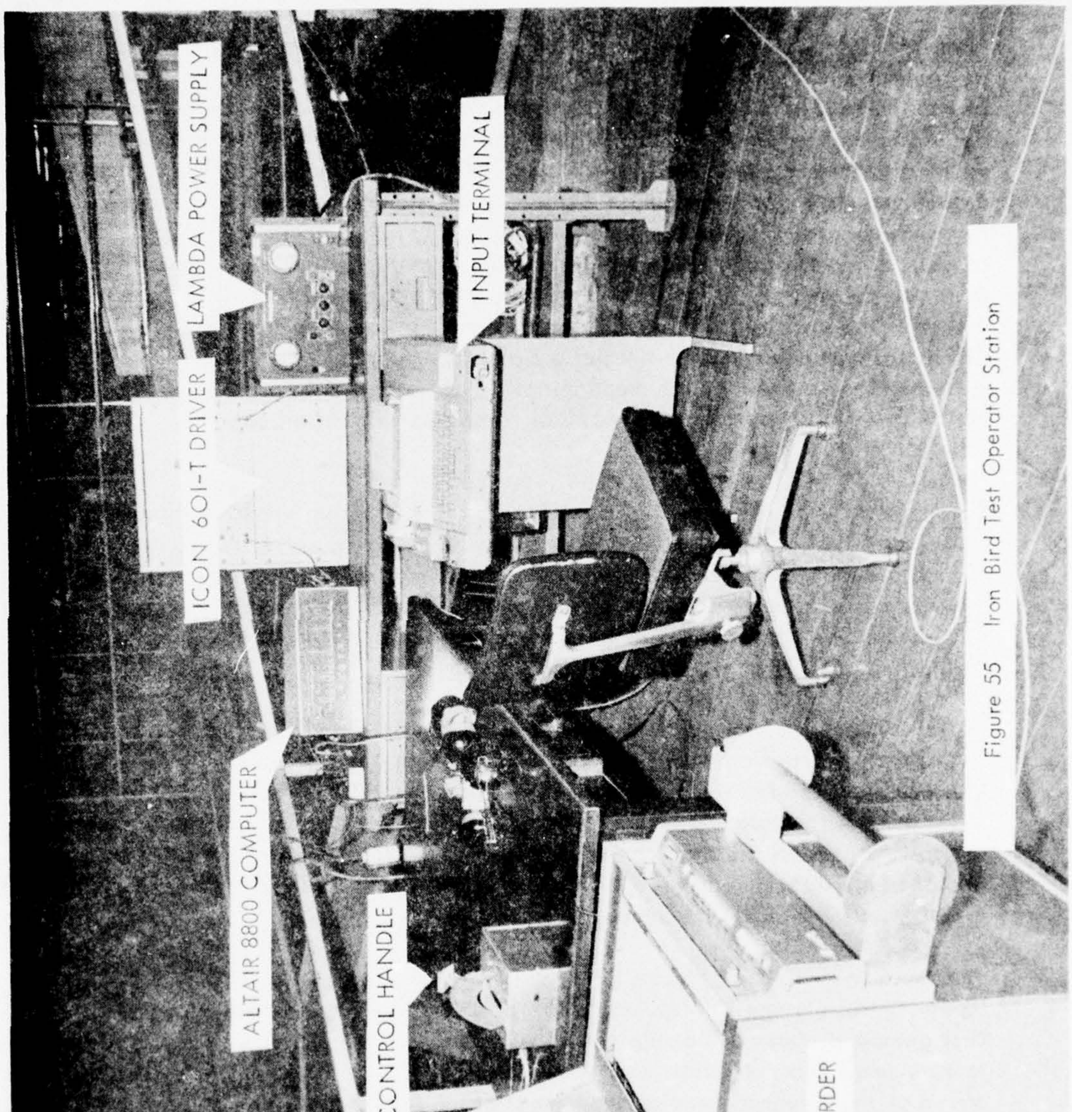


Figure 55 Iron Bird Test Operator Station

The test results are shown in a series of brush recordings to show the dynamic response of hydraulic pressures, flap input control, and output position. Flow rate can be used as a measure of system speed using the directly proportional speed/flow relationship of the fixed displacement EHPM motor. A flow of 27 gpm is equivalent to 4000 rpm of the EHPM. The flow rate curve also identifies the acceleration/deceleration characteristics.

Ground Cart Operating Runs with No Deceleration Program - The first test runs on the Iron Bird were accomplished with the hydraulic power from a hydraulic ground cart. For these runs, the software program did not provide a ramped deceleration. When the input and output encoders matched, the EPM was stopped without decreasing speed with a ramp. The flap system was actuated a number of cycles and data was recorded on a brush recorder. Performance is shown in Figure 56. Although the system was programmed for only 50% speed the pressure transients are noticeable, particularly on deceleration. Stopping was very positive causing a hard transient throughout the mechanical flap drive system. The deceleration characteristic shown on the flow trace is caused by the EHPM valve feedback after the pulse motor has stopped. This rate of deceleration is perhaps too severe for the mechanical drive system. When the decel program was added the pressure and mechanical transients were eliminated.

Engine Pump Operating Runs with a Deceleration Program - With hydraulic power delivered from the Iron Bird engine pumps and using a software program which included a decel ramp the flap system was operated through several extend and retract cycles and data was recorded. Cycles were accomplished with no load, with partial, and with maximum loads. The system was stopped and started at various positions of the load.

In the initial tests, maximum speed could not be obtained when operating with the Iron Bird hydraulic power system. Investigation determined that a high return pressure and a design characteristic of the prototype EHPM generated high thrust loads on the translator nut bearings and higher torque demands on the EPM. The return flow was routed directly to the reservoir reducing the return pressure. With this change 100% speed (4000 RPM of the EHPM) when extending the flaps and 92% speed when retracting the flaps was achieved.

The brush recordings of Figure 57 show the excellent response of the system to input control changes. Mechanical functioning was very smooth and quiet. At first glance it appears that the control is erratic, however, the system is responding to very small short duration input control changes. In the first few cycles the flaps arrive at the selected position before obtaining full speed. The acceleration ramp is interrupted and the deceleration ramp slows the unit to a stop at the selected position. Other control inputs are followed by smooth output control for the extend and retract modes of flap operation. For the selected recordings, the flap travel was controlled between 75% and 100% where the flap loads change from zero to maximum. At the center of the recording the flaps are 100% extended and the motor pressure is about 2600 psi differential, close to the design point at rated speed. On the right

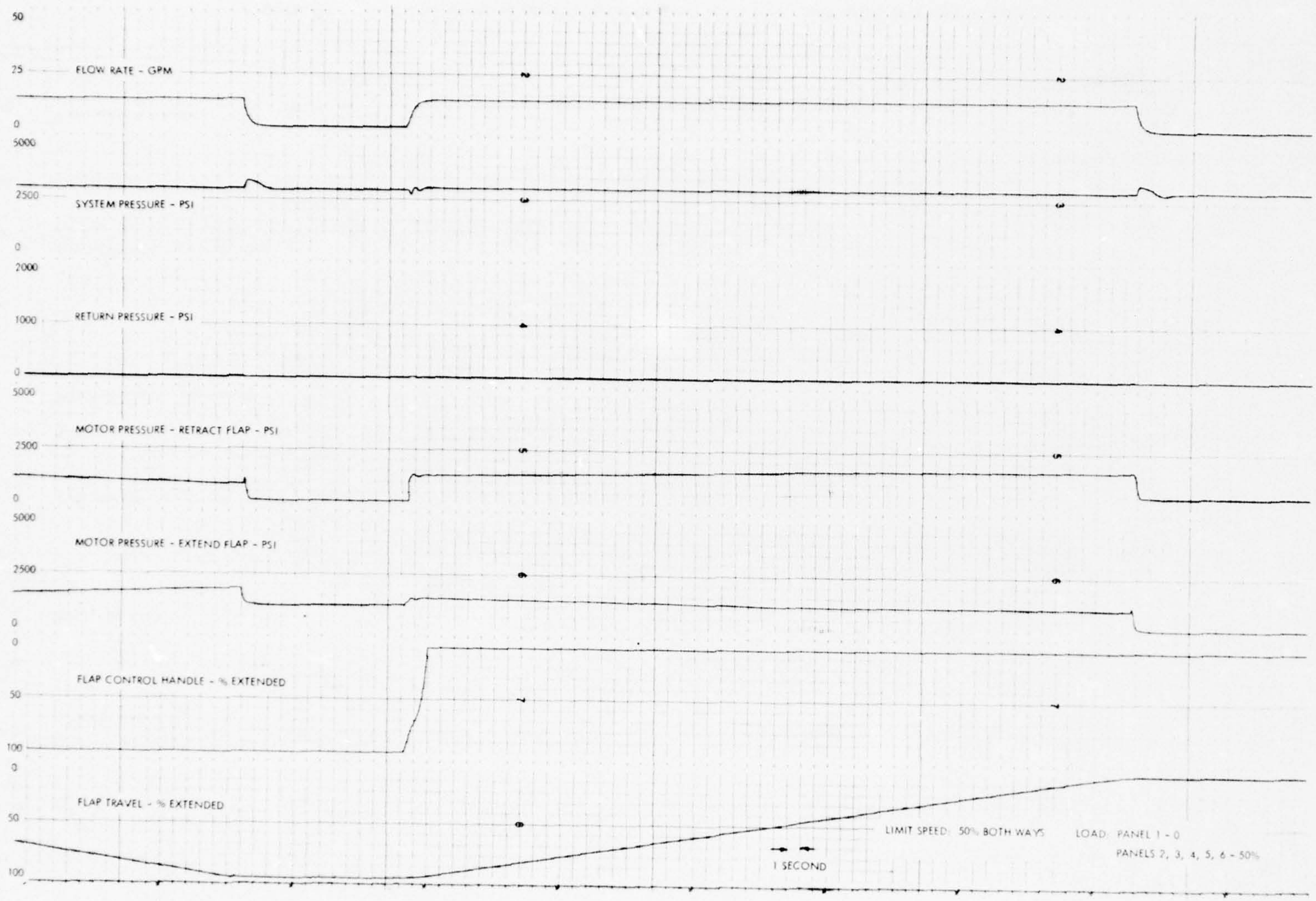


Figure 56 EHPM Performance on C-5 Iron Bird with Hydraulic Ground Cart with no Decel Program

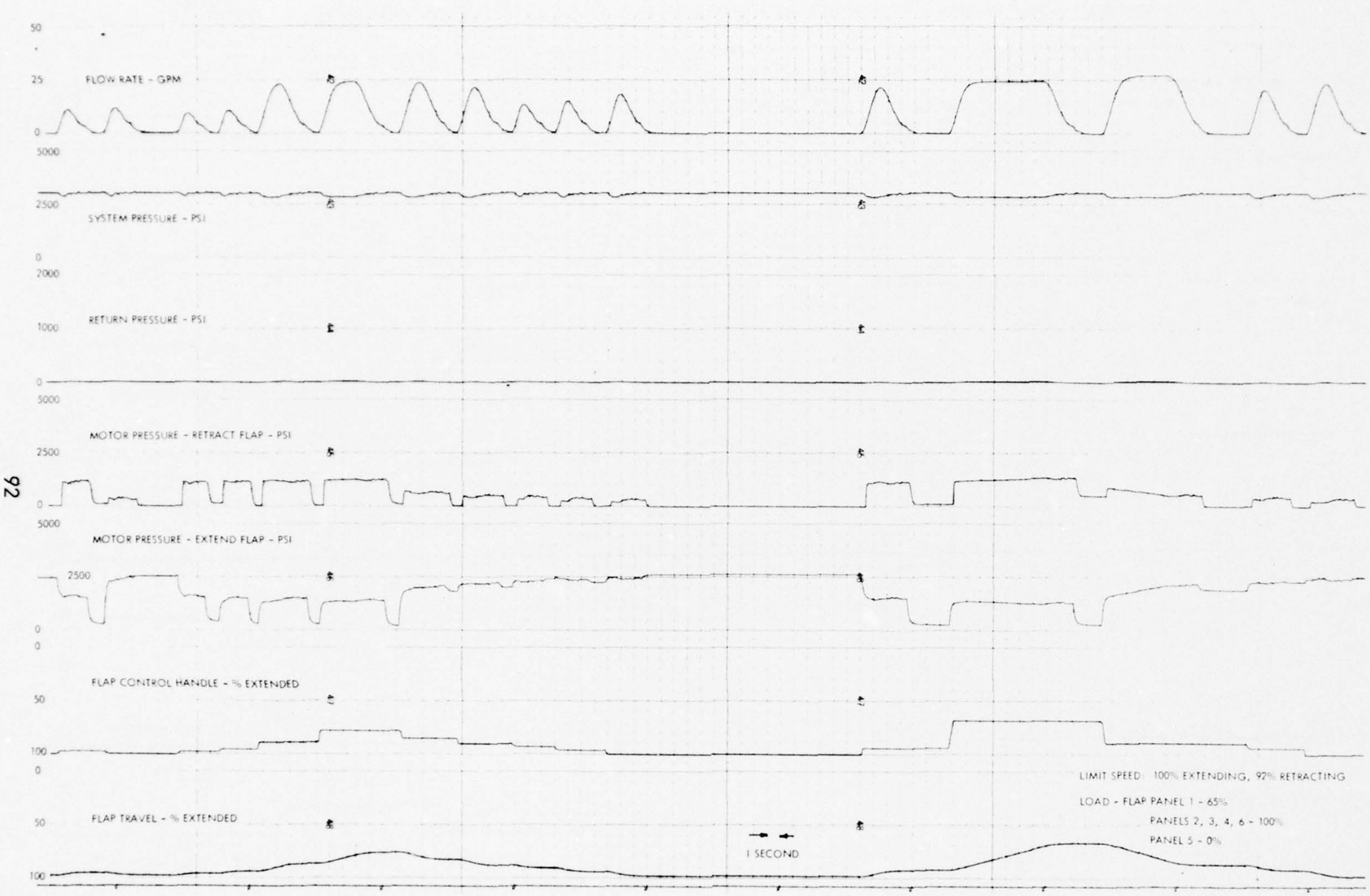


Figure 57 EHPM Start-Stop Performance on C-5 Iron Bird with Maximum Loading

side of the recording a 92% retract speed (25 gpm) and a 100% extend speed (27 gpm) was obtained. It should be noted that the programmed acceleration/deceleration ramps have eliminated pressure surges in the hydraulic system and EHPM motor and that the pressure changes occur smoothly as a result of system loads. The EHPM translates smoothly from running to stall, and the stall pressures are proportional to load. At approximately 75% flap travel the load is zero and null pressures are below 500 psi. At 100% flap travel the differential pressure is 2600 psi holding maximum load. The programmed EHPM flap drive system started and stopped smoothly for all loads and modes of operation of the input control handle, including reversal while in motion, before and after reaching limit speed and when taking a single encoder step (about 1250 pulses). A normal cycle flap operation is shown in Figure 58. The flap drive system was loaded by the Iron Bird flap actuation load system. The extend time at 100% speed was 28 seconds the retract time at 92% speed was 30 seconds.

Some mechanical problems were encountered during this phase of the testing. The problems generally resulted in jamming of the nut/screw translator drive assembly, a known deficiency of the prototype EHPM design. The jamming was caused by either (1) a malfunctioning of a torque limiter in flap panel #5 actuation system or (2) running into the stops during program trials and changes. Minor disassembly or torqueing the EPM input shaft corrected the jamming and testing continued. This problem can be corrected by non-jamming stops in later EHPM's.

Response to Inadequate Hydraulic Supply - Two tests were conducted to evaluate the behavior of the system when hydraulic supply was not adequate causing system pressure to drop. In the first of these tests, system supply was adjusted by pump speed so that system capacity was only slightly more than adequate to operate the EHPM. A fixed by-pass flow of 9 gpm (limited by a flow regulator) was suddenly applied while the EHPM was operating. The by-pass load simulated the operation of some other load in the system such that the total flow required exceeded the capacity of the system. In the second test the engine pump was operated at decreasing speed so as to decrease its capacity. The EHPM was operated while capacity was decreasing to record operation at various levels of inadequate flow rate.

When operation of some other device in the system causes system capacity to be exceeded, system pressure will drop below normal and can under some conditions drop below flap brake release pressure. This will cause the brake to cycle on and off or drag. Either case is unsatisfactory and the control system must prevent this mode of operation. Therefore, the program reduces the EHPM speed when system pressure is below 1500 psi and causes it to stop when the pressure is below 1200 psi. The brake is applied by shutting off pressure to the system via a shut-off isolation valve when system pressure is below 1200 psi. To prevent uncontrolled cycling a 2-second delay is programmed between the shut down and restart due to pressure rising above 1200 psi. Performance for this mode of operation is typified by the recorded data for one run shown in Figure 59.

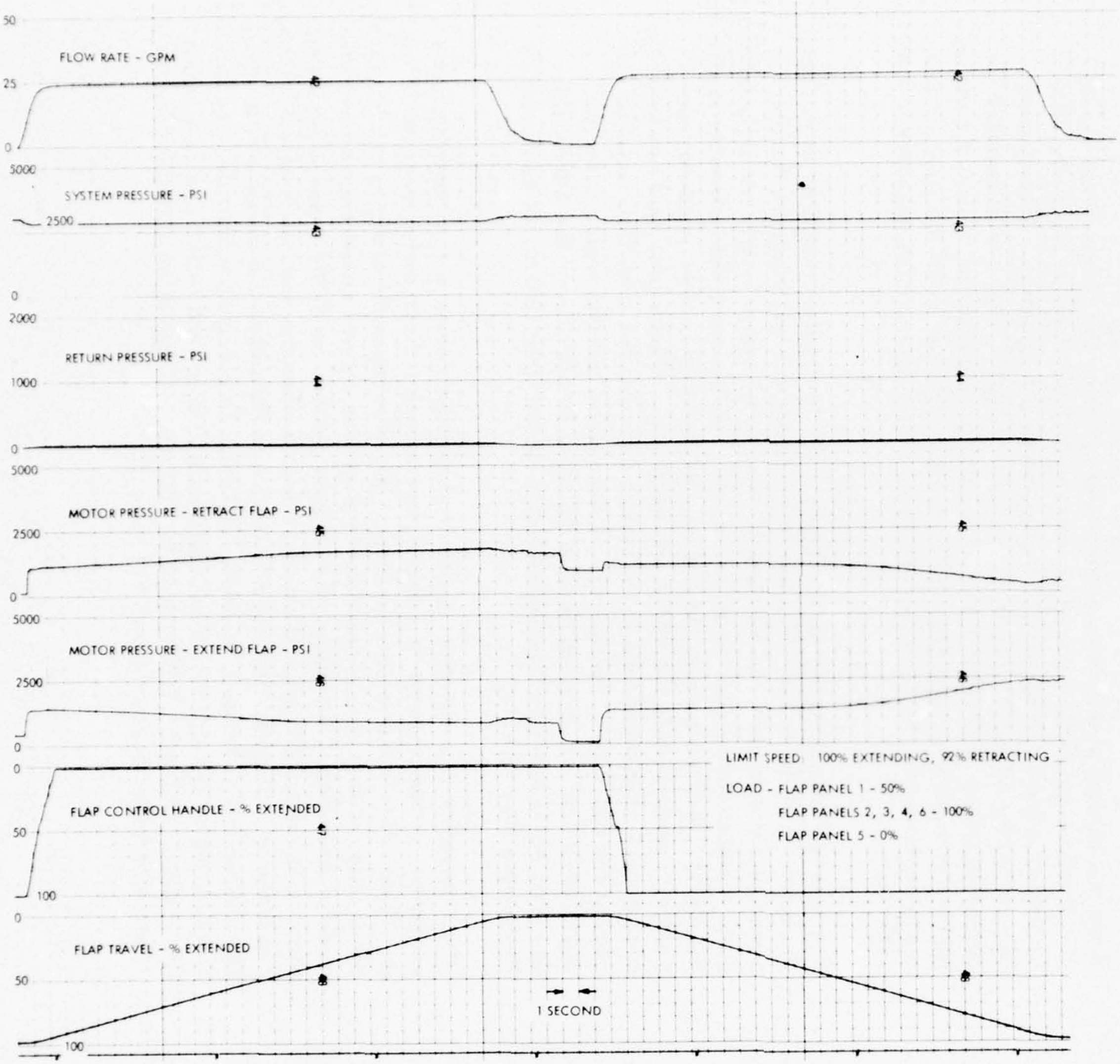


Figure 58 EHPM Full Cycle Performance on C-5 Iron Bird with Partial Loading

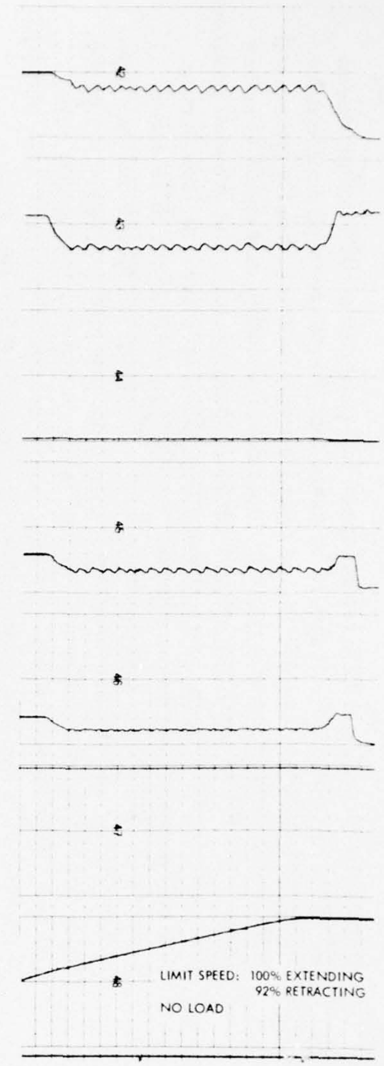


Figure 59 Response of EHPM to Load Exceeding System Capacity

The pulsing nature of flow and pressure reflects the control algorithm used and the resolution available with the 8-bit analog to digital converter (ADC) used to digitize the signal from the pressure transducer. The algorithm causes the speed to be stepped down one notch for each program cycle if pressure is below set point (1500 psi) and to be stepped up one notch per program cycle if pressure is above set point. Although the flap travel data of Figure 59 does not reflect this pulsing, the EHPM did pulse. An approach to smoothing this pulsation is to increase the time between speed changes if the pressure is near the set point. For example, if the pressure is above 2000 psi the time between speed changes could be the normal .015 ms and if it is below 2000 psi it could be .15 ms. Selection of this value by trial should select the lowest value which will give acceptably smooth operation. Higher resolution of the pressure would allow adjusting speed within a narrower pressure range.

To evaluate this pulsation throughout the full range of flow inadequacy the speed of the engine pump was gradually decreased until flow was about 5 gpm. The recorded data for this run is shown in Figure 60. The pulsing nature of flow and pressure is evident, however the program did control the speed at the available reduced flow.

Further attempts to smooth this pulsation were limited by time.

Failure Mode Tests - Hydraulic failure mode testing was conducted by decreasing system pressure to less than 1200 psi to demonstrate the shut-off feature in the program. Pressure was decreased by adjustment of the ground cart pump compensator.

Electrical failure modes were accomplished using the durability test arrangement. With the unit running, electrical power to the EPM was interrupted. The test was repeated several times. The unit stopped each time power was interrupted.

It is concluded that the EHPM concept is compatible with a typical aircraft hydraulic system, that pressure surges can be minimized, and that the concept lends itself readily to optimization of performance by adjustments to control parameters in the software program. It is also concluded that the prototype EHPM is inadequate in two respects. (1) The drive screw jams when the valve reaches its travel limit. (2) The effect of high system return pressure and the limited torque capacity of the EPM.

Durability Tests

The durability test set-up is shown in Figure 61 and is schematically the same as Figure 45. A computer program was written to simulate flap operation and to cycle the EHPM a minimum of 5000 cycles in accordance with the following schedule.

Normal Landing Cycles - Definition of one cycle: Starting at 0° flap angle, extend to takeoff and approach position (21 seconds), stop (wait 2 seconds), extend to landing position (5 seconds), stop (wait 1 second), return to 0° flap angle (26 seconds), stop (wait 2 seconds). Total cycle time 56 seconds.

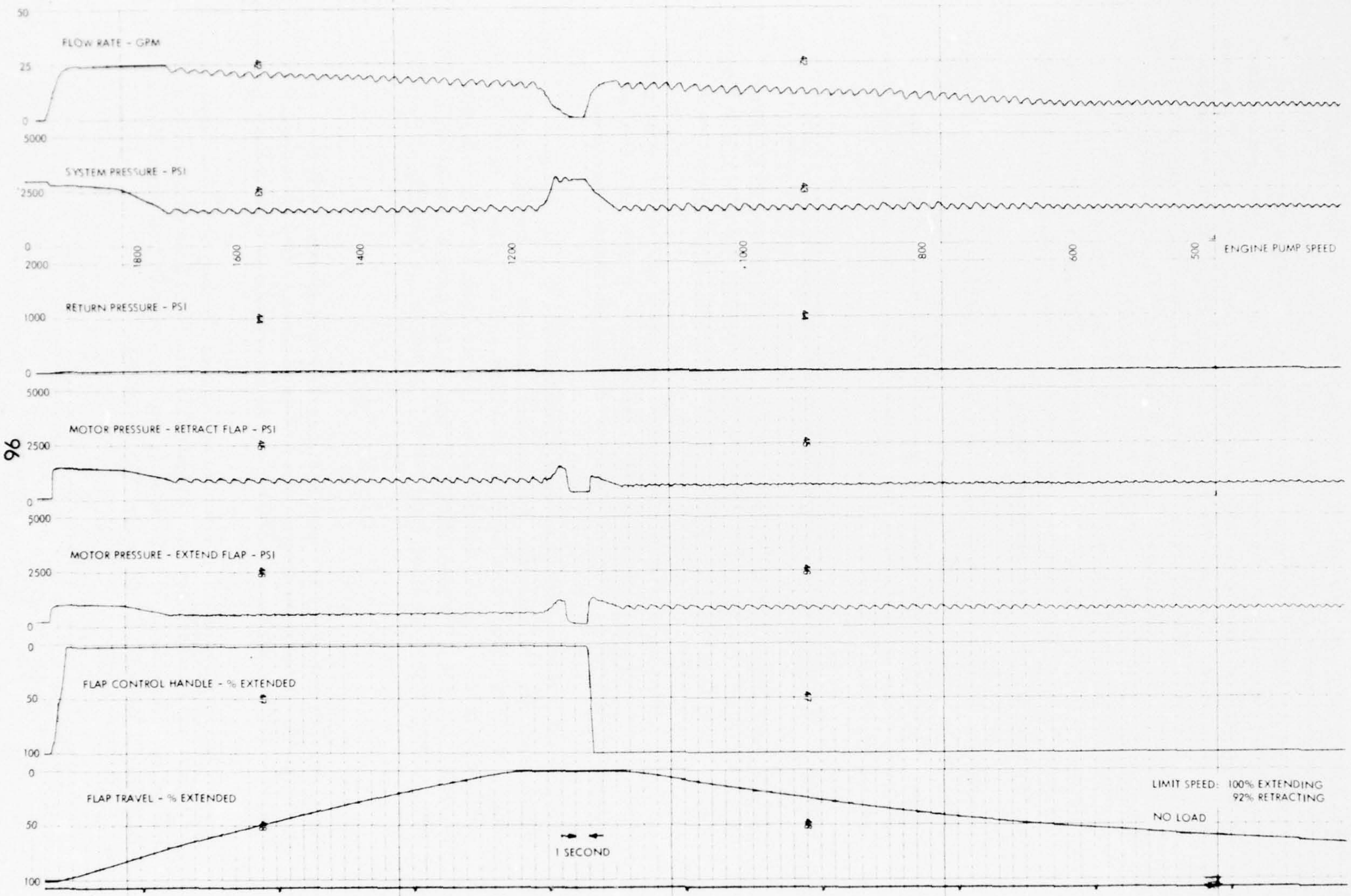


Figure 60 Evaluation of EHPM at Various Levels of Flow Inadequacy

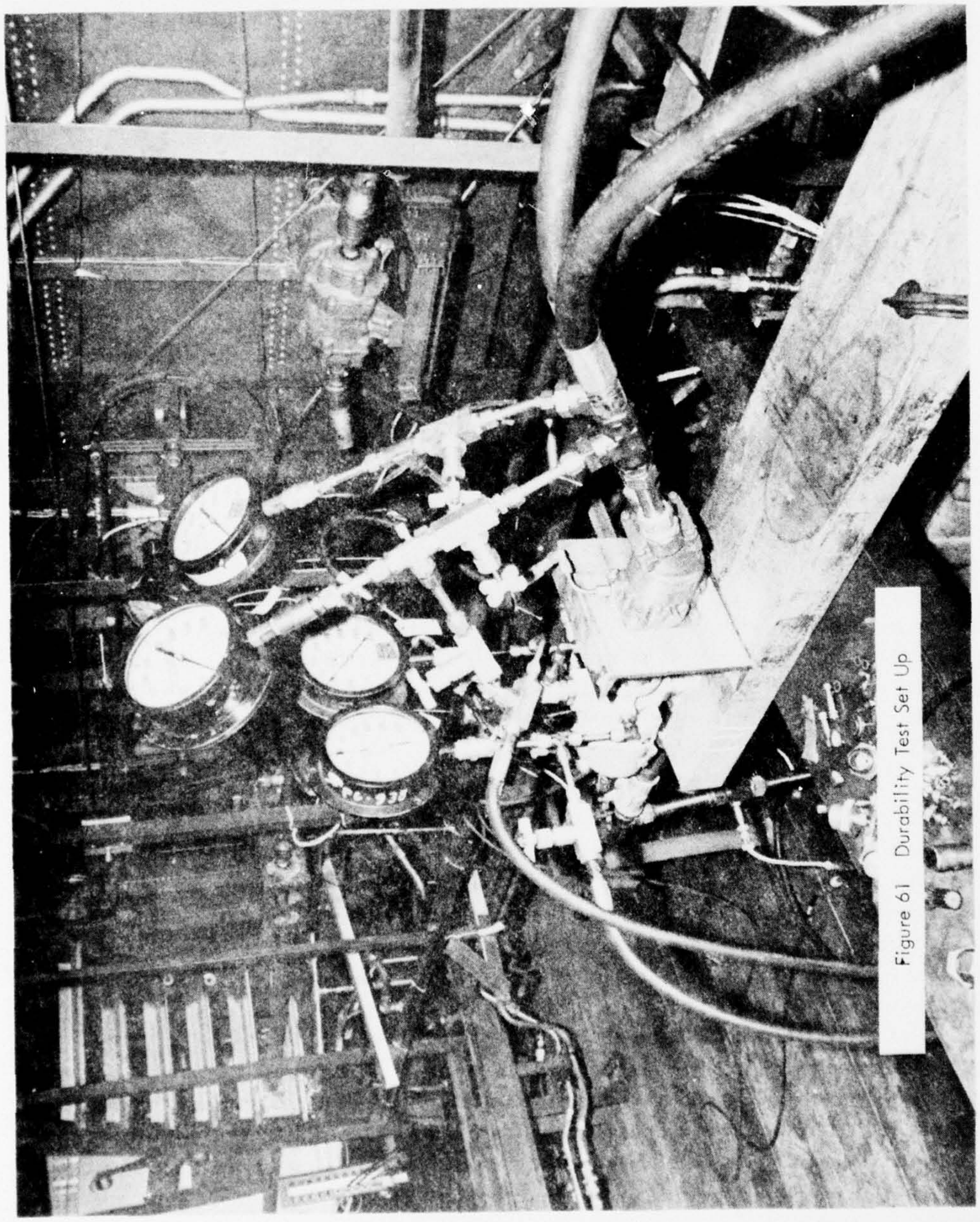


Figure 61 Durability Test Set Up

Load	Cycles Required	Cycles Completed
0	2500	2481
65%	1250	1287
100%	416	472
Sub Total	4166	4240

Touch and Go Cycles - Definition of one cycle: Starting with the flaps at the takeoff and approach position, extend to landing position (3 seconds), stop (wait 2 seconds), return to the takeoff and approach position (3 seconds), stop (wait 2 seconds). Total cycle time 10 seconds.

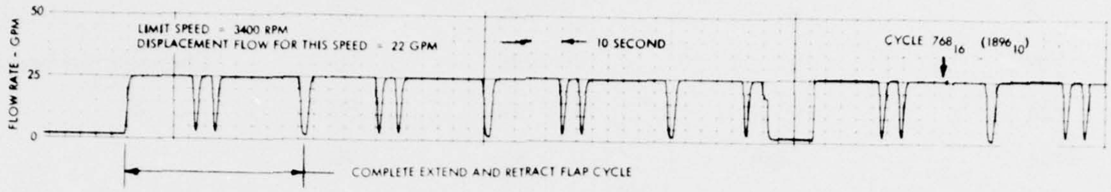
Load	Cycles Required	Cycles Completed
0	667	709
100%	167	240
Sub Total	834	949
Total	5000	5189

The unit was cycled as shown in Figure 62.

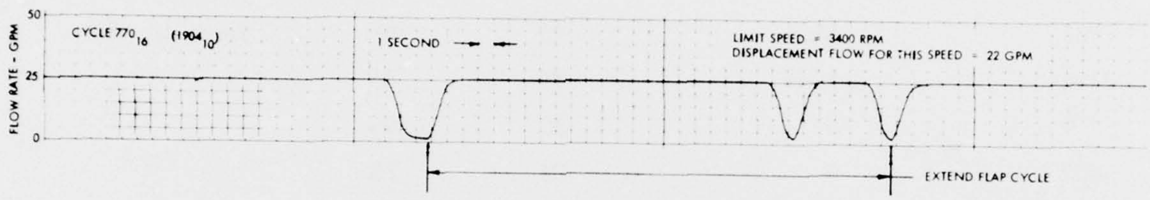
At the startup of each day's testing, several cycles were required before the EHPM would consistently reach limit speed. A procedure to run at some intermediate speed until the warmer system oil was circulated through the unit was established prior to running on the programmed durability cycle tests. The oil temperature was maintained at 100°F through 680 durability cycles and then raised to 150°F for the remaining cycles. After 250 no load, full cycles the unit would not reach limit speed even after warm-up. The unit was disassembled and the outer bearing of the translator nut assembly was found to be rough. The bearing was replaced with a Fafnir 9103K bearing and testing was continued. Later the electric pulse motor was also changed due to a rough bearing in the EPM. Modifications to the return lines were also made to reduce return pressure from 100 psi at 85% to 65 psi at 92% speed. During the tests return, null, and differential ($C_1 - C_2$) pressures increased and internal leakage became excessive. To allow continued operation, the limit speed had to be dropped as shown in Figure 62. The limit speed was reduced to 75% (3000 rpm) prior to the completion of full flap cycle operation at 3602 cycles. Flow recordings taken at various times are shown in Figure 63. Flow recordings were not taken until maloperation of the unit occurred and high internal leakage was suspected. Figure 63 shows that leakage flow was about 2 gpm at cycle number 1904 and about 8 gpm at cycle 5076 and that this leakage is present while stopped and at speed and that internal leakage was again low after repair and reassembly.

Cycles Normal	Load	Speed - % of 4000 RPM Extend	Retract	Remarks
0	0	100	92	
249	0	100	90	Changed nut bearing
958	0	95	90	
1461	Max	95	90	
1538	Max	92	90	
1560	0	90	90	Change EPM
1597	Max	85	85	
1711	0	85	85	System return change
1933	Max	85	85	Trace 4a, 4b
2192	Med	85	85	
3479	0	85	85	
3602	0	75	75	
Touch & Go				
4240	Max	75	75	
4480	Med	75	75	
5189 Complete				Trace 4c

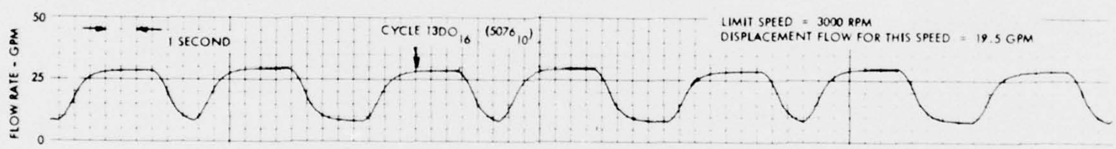
Figure 62 Summary of Durability Cycling



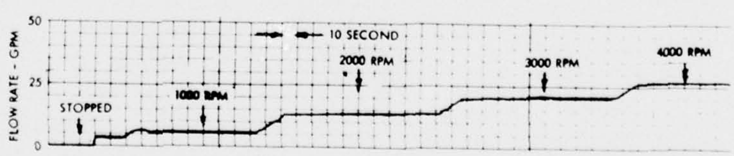
TYPICAL COMPLETE FLAP CYCLES



TYPICAL EXTEND FLAP CYCLE WITH FASTER RECORDING



TYPICAL TOUCH AND GO CYCLES HAVING
HIGH INTERNAL LEAKAGE



WITH DC MOTOR DRIVE AFTER
REPAIR SHOWING LOW INTERNAL LEAKAGE

Figure 63 Durability Flow Tests

After completion of the durability cycles the EPM was replaced with the D.C. motor to run the valve performance tests. During the drag torque tests, when running in the clockwise (CW) direction, a failure occurred that caused the speed to increase followed by a bump and stopping of the unit. Following the failure, a bad seal leak at the EPM gearbox was present when running and the unit would not start with 17 amps current in the CCW direction. A tear-down inspection revealed that the sleeve retaining nuts had loosened allowing the sleeve to move enough for "O" rings to fail - resulting in high internal leakage which prevented normal performance. A retaining pin in the sleeve had also failed, apparently as a result of the valve bottoming out. Two material voids on the spool, shown in Figure 64 were found that appear to be where pieces of chromeplate flaked off. A score was found on the sleeve bore at a position where the flaking occurred on the spool, indicating that the flaking caused the score. Removal of the score by lapping allowed reassembly and completion of the drag torque and leakage tests. The lapping was not in the area of a leakage path. Internal leakage was found to be less than at the beginning of the test program. The disassembled unit is shown in Figure 65 and 66.

Analysis of the failures leads to the following conclusions:

- o The effect of return pressure - increased return line pressure causes the translator nut load and torque to increase until the EPM driving torque is exceeded and the unit stalls. The prototype design allows return pressure to act on the full area of the seal mate, Figure 34, instead of the face seal area only and increases the bearing load by a factor of four. A design change to limit travel of the seal mate would improve the design and reduce the effect of return pressure.

- o Internal leakage caused by sleeve "O" ring seal failures - The progressively deteriorating condition of the EHPM was caused by the increased internal leakage and seal failures. The valve sleeve retaining nuts were loose on disassembly, allowing the sleeve to move and the seals to fail where high differential pressure was present. The retaining nuts are threaded and were loosened by alternate torques transmitted through the sleeve when the translator nut/screw assembly was jammed or on spool/sleeve momentary seizure during the drag torque test following the durability runs. This torque to sleeve condition is not present in purely translational spool/sleeve assemblies but is inherent in the rotating spool concept of the EHPM. An improved method of sleeve retention will fix this problem.

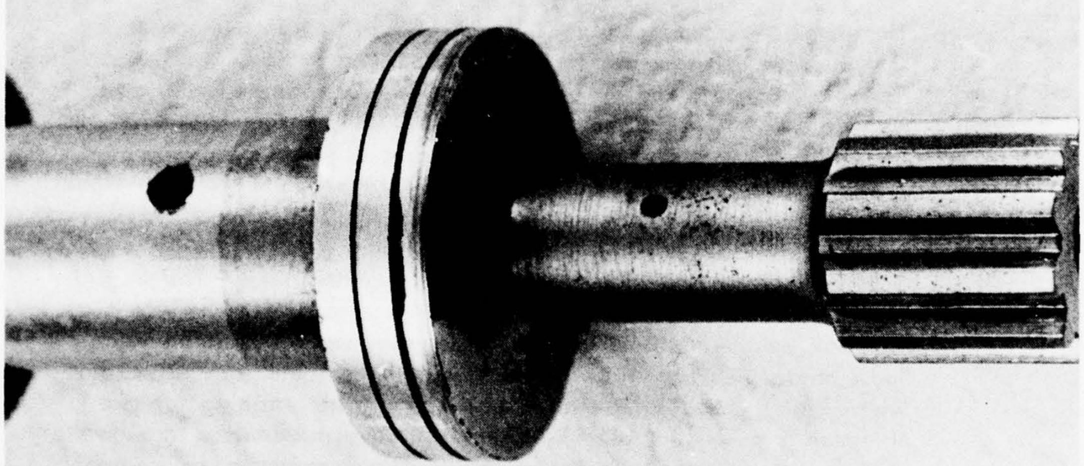


Figure 64 The Valve Spool Showing Material Voids

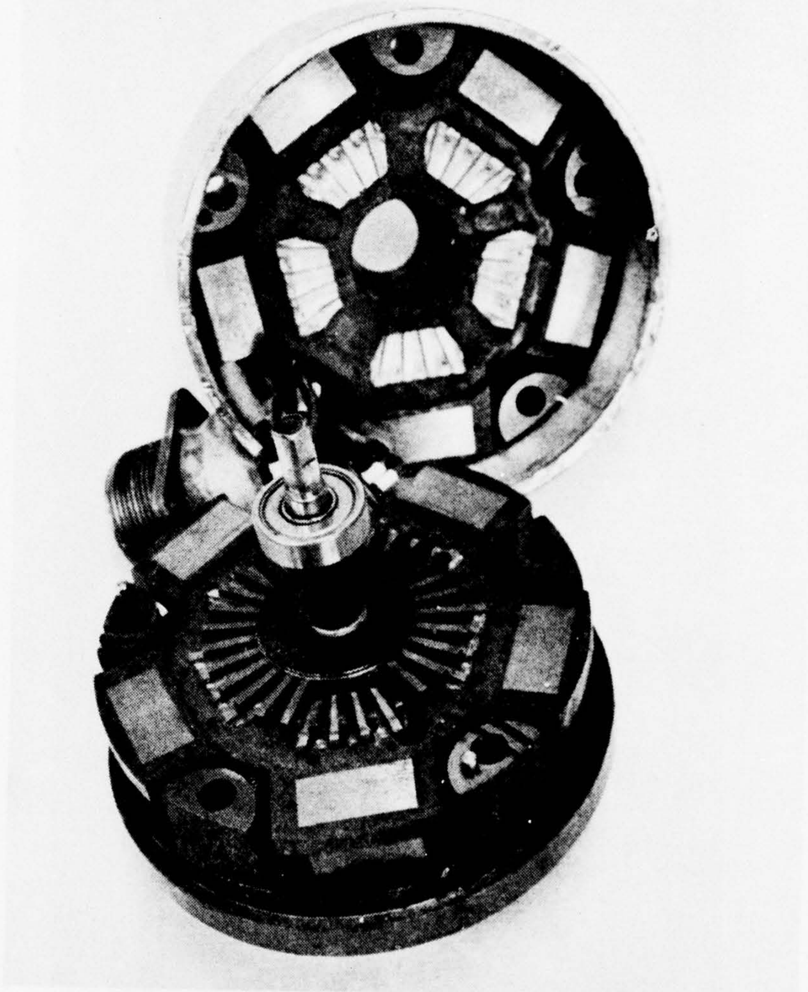


Figure 65 The EPM Disassembled

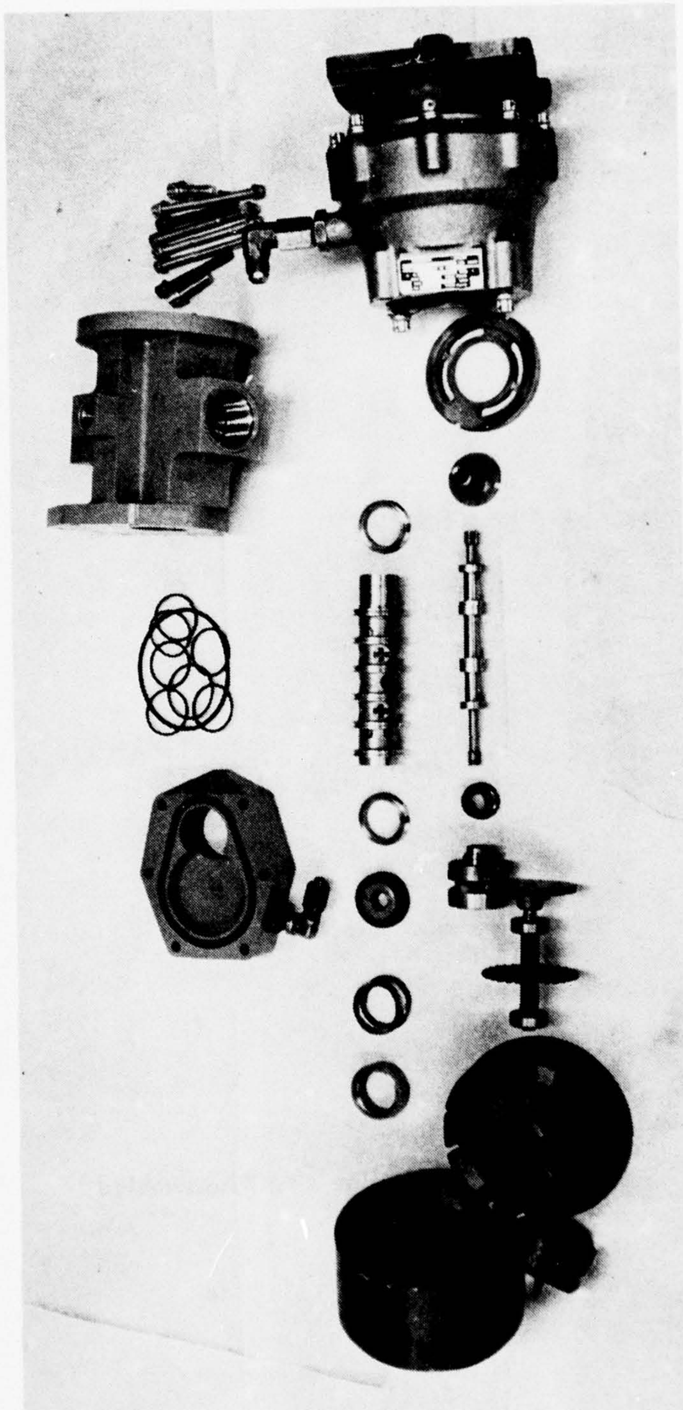


Figure 66   The EHPM Disassembled

- Spool/sleeve wear or erosion - In the limited durability test, no deterioration or wear of the spool/sleeve assembly was evident on teardown or in the final valve performance tests. It is thought that the momentary seizure during the final valve performance test was caused by separation of the chrome on the repaired spool used in the final assembly.

- Performance of the unit after repair and assembly - In respect to internal leakage and null pressure performance after repair indicates that some "O" ring damage may have been present at the beginning of the test program.

- Valve Drag Torque - The valve drag torque measurements shown in Figure 67 indicate a change in torque above 2000 rpm in the CW direction. The original CW torque test shows a drop in torque above 2000 rpm. The later test shows a rise in torque. The rise in torque over the earlier test may be due to the score in the sleeve. The spool is running in about the same position as the score.

It is concluded that wear of the spool and sleeve was negligible; that research for a spool sleeve material and gap combination not dependent on a plating would be a worthwhile effort; and that other failures which occurred are preventable in future designs via state-of-the-art detail techniques.

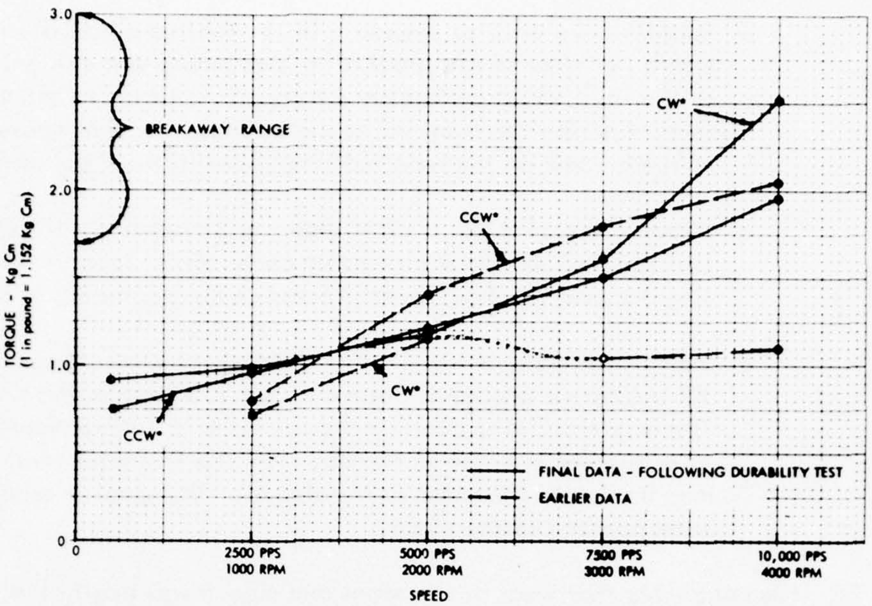


Figure 67 Valve Drag Torques

SECTION VII

CONCLUSIONS AND RECOMMENDATIONS

CONCLUSIONS

State-of-the-Art and Available Hardware - EPM technology and input system technology including the digital computer are adequately developed to support the use of the EHPM concept for aircraft to be designed in the next decade. EHPM's presently in use are not appropriate for aircraft use, but their successful use for machine tools is a significant substantiation for the basic concept. Aircraft EHPM's will not require the high resolution of machine tool units, but will require higher speeds so as to minimize weight.

Component Analysis - The hydraulic motor needs to operate at a higher speed than the EPM so that the EPM can operate on the higher portion of its torque curve. The resolution of the aircraft EHPM can be several orders of magnitude less than the machine tool unit so this EPM speed reduction is achievable. All loads including inertia and viscosity reflected to the EPM need to be minimized and need to be dependable. The size and weight of the EPM needs to be reduced. These conclusions lead to a singular conclusion; namely, the EPM should operate a pilot section which in turn hydraulically positions the main spool. The pilot section is constructed as small as physically feasible and arranged so that pressure loads are balanced and viscous and inertia loads are nil.

Open Versus Closed Loop - The arrangement used for the Iron Bird tests where the input is an absolute binary encoder and an identical encoder is driven by the output of the system is considered to be an excellent approach for aircraft utility and secondary flight control subsystems. The system operates open loop for acceleration and speed control but closed loop for position control. As was done, pressure signals close the loop when necessary. Open loop may be useful for some applications which involve loads that will never exceed design values or where failure is of little consequence. Open loop can be used only if the EHPM always has the capability of achieving the motion corresponding to the EPM motion. If the power demand exceeds the power capability the valve will bottom out, and the EPM will stop. With the arrangement used in the flap drive system the output encoder is examined to determine if it has stopped, and if it has the EHPM is restarted. If the unit is arranged so that bottoming out does not jamb it, so that the only result is to stop the EPM, then the flap drive systems offers a good solution. If the EPM is used only within its start-stop range of speeds and limit switches are provided, open loop could be used between the limits. It is concluded, however, that for most applications the loop should be closed as was done in the flap drive system. The hardware used for closing the loop - i.e. encoders vs. other types of position sensors needs further study to assure minimum cost.

Prototype EHPM Design - Non-jamming thread stops are needed on the translator. The valve sleeve retaining nuts need positive locking, and the sleeve should have a torque tie to the body other than the retaining nuts. The face mating seal needs support to the body so that return pressure will not force it against the nut bearings. The vent passages in the spool should vent the spool ends to the hydraulic motor case but not to the valve return. The EPM torque is not adequate for the design.

RECOMMENDATIONS

Additional Testing - Several areas of additional testing using the equipment developed in this program can add economically to EHPM technology. Spool and sleeve material combinations and clearances need to be evaluated. The chrome plated spool used in this program apparently caused a failure and if unplated material can be used the plating-failure mode can be eliminated. It is also feasible to modify the design of the new sleeve and spool specimens so that return pressure will not load up the nut bearings. The new specimens should be subjected to the durability test.

Additional programming with operational evaluation to smooth out the pressure control functions and to evaluate acceleration limits should be performed. The latter item involves programming the computer so that at each speed the maximum acceptable acceleration to the next step is found. The data obtained allows selection of an accelerations ramp with a known margin.

Conceptual Design - Conceptual designs using a pilot section and a speed reduction between the hydraulic motor and the valve and overcoming the faults discovered in this program should be developed. Layout studies already accomplished indicate a high probability for success of this effort. This effort should investigate other alternatives.

Black Box Definitions - A black box which can control any subsystem or parts of various subsystems and whose functions can be standardized so that only the program has to be different for different applications appears to be a realistic concept. Studies leading to the definition of the functional requirements for such a control should be pursued.

Linear Actuator - The EHPM concept is applicable to linear actuator control and the approach appears to be a viable alternative for thrust reverser control and cargo ramp control where multiple actuators must operate synchronized. A conceptual design study for such a device based on the C-5 thrust reverser requirement should be accomplished.

Application Studies - Preliminary design studies for applications where the pay-off appears to be significant should be performed. The thrust reverser application, ala C-5 requirement, is especially appropriate. The basic requirement is applicable to future systems, and a cursory analysis of the C-5 system indicates a high probability for a good pay-off.

Systems Studies - The basic system approach used for the Iron Bird tests is believed to typify the control system of the future. The selected elements used in this program, their concepts and their packaging do not, however, portray the technology which is to be used in a production system. The trend, though, in the technology for these elements is obviously leading to costs, size, weight, and performance that will make the approach irresistable. Studies need to be made in the areas of packaging, i.e. where to put the various functions; position feedback elements, i.e. what concept is

optimum to perform the encoder functions; computer capability requirements, i.e. how fast and how much memory; and programming techniques, i.e. how to make the system react smoothly to the various forcing functions.

APPENDIX A

GLOSSARY OF TERMS

This section is divided into the following sections: EHPM, EPM and Input Control System. Most of this information was obtained from References 1 and 2.

Electrohydraulic Pulse Motor - EHPM

An electrohydraulic pulse motor, referred to in the literature by two acronyms, EHPM and EHSM, is a power amplifier which, in combination with its electronic control, transforms a stream of low power electrical pulses into high power rotary motion. Above some low speed the rotary motion is smooth and not pulsatory. The displacement of the hydraulic motor shaft is proportional to the number of pulses delivered; the speed is proportional to pulse rate; and the acceleration is proportional to the change in pulse rate. All of these can be controlled in open loop. The EHPM consists of an EPM which is mechanically connected to a four-way valve such that a displacement of the EPM causes the valve to translate and deliver flow to the hydraulic motor whose rotation causes the valve displacement to be cancelled. The arrangement causes the hydraulic motor to follow the EPM. Some terms which are used in EHPM technology are defined as follows:

> <u>Translator Driver</u> - this term has been used twice in the current program. In one case it is the threaded nut within the EHPM which causes the spool to translate when the EPM rotates and in the other it is the Icon 601-T electronic package which includes a 28-volt power supply, a logic circuit to convert pulses to sequential energization of the five power amplifiers which supply the higher electrical power to the five EPM coils.
>
> <u>Electrical Pulse Motor</u> - the electrical pulse motor referred to in the literature by two acronyms, EPM and ESM, is the input device of the EHPM and because of its important role its description and the terms used in its technology are presented under a separate heading which follows next in this glossary.

Electrical Pulse Motor (EPM)

<u>Accuracy</u> - The step positions in a motor are determined by dimensions of the parts and assemblies built into it during manufacturing. Tolerances in these dimensions result in tolerances in step positions from the nominally correct locations. This is termed <u>accuracy</u> and is expressed as a maximum angular dimension. For example, a maximum error of 0.25 degree in a 24-step-per-revolution (15 degrees/step) could be given as a ±1.67% accuracy. Accuracies of commercially available motors range from ±5% to as low as ±1%.

These built-in position errors are systematic, or repeatable, for any given motor, and noncumulative. They occur under all load conditions. Loads involving significant internal friction are subject to an additional nonrepeatable positional error. This more-or-less random error is unrelated to the built-in motor accuracy, although the two types of error are additive in the system. This error is also noncumulative. The value of this error can be obtained from a plot of the holding torque.

Efficiency - The power out of the stepping motor divided by the power into the driver. This ranges from 0, when the motor is just holding, to 10% at best, when the motor is running. Steppers are positioning devices, and are very poor at energy conversion.

Note that this low efficiency applies to the stepper only; the efficiency of the hydraulic motor is far higher. Also, the low efficiency of the stepper reduces the efficiency of the electrohydraulic system only slightly because the power input into the stepper is a small fraction of 1 hp, while the hydraulic motor may involve many horsepower.

Inductance - All motor windings have self-inductance and many motors also have mutual inductance. Mutual inductance is essentially nonexistent in multiple-stack VR motors. When the term inductance is used, it commonly means the self-inductance. The inductance varies with the rotor position and current in the winding. The inductance is highest with rotor and stator teeth aligned and no current flowing in the winding.

Internal Inertia - The internal inertia of the motor is useful in evaluating the effect of load inertia on motor performance. However, due to the unique time-variant and nonlinear nature of motor torque, the conventional torque-to-inertia or torque-squared-to-inertia ratios are of questionable value for other than rough comparison purposes. Rotor inertia includes the inertias of all rotating components. When selecting a motor, the rotor inertia must be included in the calculations.

Minimum-Response-Time Curves - The minimum-response-time curves are much more useful than pull-in torque curves. They indicate the minimum transit time to travel the indicated distance, given in degrees for each curve, with an inertial load (no friction) rigidly coupled to the motor shaft. The values of the load inertia are given along the abscissa on a logarithmic scale. The time in seconds/cycle covers the start-from-rest through return-to-rest at the end of travel. The curves for short motions are obtained with some form of damping. For long motions, the transit time is determined by accelerating and decelerating the motor. The last pulses are timed to cause the motor to have zero velocity at the time it arrives at the final position. If the motor has some velocity as it reaches final position, it will ring and additional settling time is required.

Resolution - Resolution is expressed as the number of steps per revolution, or step angle in degrees. This is an unalterable, built-in characteristic of stepper motors.

The VR and PM motors are available in the range of 1 to 500 steps/revolution. The nutating disc and flexspline types, from 400 to 2000 steps/revolution.

Resonance - EPM's have a natural frequency. Stepped at their natural frequency, they will resonate and lose step. They will go through resonant speed error-free. Inertia load rigidly coupled to the motor shaft lowers the natural frequency and resonance.

Speed - EPM speeds are normally given in steps per second. Dividing the steps per second (speed) by the steps per revolution (resolution) and 60 seconds gives the shaft speed in revolutions per minute (rpm). The following types of speed are encountered in EPM technology:

- o Start-stop speed is the highest step rate to which the motor will respond without step loss during starting and stopping.

- o Slew speed is generally the fastest speed at which the motor will run unloaded with very slow acceleration.

Stepper Motor (EPM) or (ESM) - Devices, generally electromechanical, which in response to a signal input, assume a known position. The signal may contain the the necessary power to shift the load, but generally does not in the larger sizes of steppers. Four main types - variable reluctance, permanent magnet, nutating disc, and flexspline - are compared as follows:

- o Variable reluctance (VR) and permanent magnet (PM) motors both generally have a rotor and stator with teeth. Energizing a coil creates magnetic flux which aligns the teeth. Different coils cause different teeth to align and permits sequential stepping. PM types have a permanent magnet in the circuit which aids or bucks the flux induced in the coils. PM types have a detent action when electrical power to the coils is removed.

- o Nutating disc type steppers have a wobble disc with two sets of low angle bevel gears connecting the disc to the output shaft. Solenoids positioned in a circle sequentially attract the disc, rotating the shaft.

- o Flexspline motors use a thin wall flexible cylinder with minute gear teeth on the outside and thin layers of coiled magnetic material on the inside. Radially positioned solenoids attract the magnetic material, distorting the cylinder into an oval shape. The gear teeth at the high points engage a rigid internal spline. Sequentially energizing the solenoids moves the points of contact. A 2-tooth difference causes the flexspline to rotate small amounts. The output shaft connects to the flexspline.

Torque - The following types of torque are encountered in EPM technology:

- o Holding torque is the maximum steady torque that can be externally applied to the shaft of an excited motor without causing continuous rotation.

- o Detent torque is the maximum torque which can be applied to a non-energized stepper before the rotor will snap out of position. Only PM motors have significant detent torque.

- o Stall torque is the maximum torque available while running at a very low speed.

- Running or pull-out torque is the maximum constant torque load against which the stepper can run error free, after it has reached speed. This is the torque shown in most manufacturer's torque vs. speed curves. This torque measurement is only slightly influenced by inertia loading.

- Pull-in or torque-to-start-without-error is the maximum torque load that the motor can pull into synchronism with a constant frequency step rate. This torque is determined by putting a fixed torque load on the motor, then giving the motor a constant frequency pulse rate. The pull-in torque is the largest pure torque load against which it can start and remain error free (that is, without any step loss). No inertia load is considered in this definition.

Windup - Stepper shafts will twist under torque loads. The VR and PM types are fairly linear, and exhibit little hysteresis.

Mechanical motors do not exactly have backlash, but the windup is not linear about zero, and there is more hysteresis.

Input System

EPM Electronics - The stepping motor drive system accepts low power IC level pulses and converts them to the proper phase format for the motor windings; a power amplifier then switches the required current through the windings. Choosing the proper drive circuit design is an important as selecting the proper motor.

Some drivers use constant current power supplies; others use constant voltage power supplies. Constant voltage types are simple and inexpensive. Constant current drivers are more complicated, but offer better performance. The characteristics of several types of drivers are as follows:

- R/L Drives - The coils of the motor have an inductive time constant which limits average coil currents and motor torque at high stepping rates. To reduce circuit time constant, an external resistor is connected in series with the motor coils, permitting an increase in power supply voltage without a corresponding increase in coil current. The net effect is to decrease the

circuit time constant and increase high speed torque. In practice, the voltage "overdrive" is increased to 10 or more times the rated motor voltage. The major disadvantage of using external resistors is the heat which must be dissipated by them and the resulting increase in package space and weight.

- Uni-Polar vs. Bi-Polar Operation - In the uni-polar one-half of the motor winding is energized alternately for each step and the other half is unenergized. The bi-polar drive operates all coils simultaneously in a push-pull mode through polarity reversal rather than through winding selection. Typically an increase of 30% in output power can be obtained from the same electrical input power to the motor with only a slight increase in circuit cost. Bi-polar R/L drive circuits do, however, require double ended or center tapped power supplies.

- Bi-Polar Chopper Drives - The chopper drive eliminates the need for external series resistors and provides even higher operating speeds requiring less than one-third of the power consumption than R/L drives. A single high voltage power supply is incorporated with a bi-polar amplifier. The voltage to the motor coil is chopped to maintain rated motor current. The chopping function is controlled by a feedback circuit which continuously senses motor current. Thus, the requirement for current limiting resistors and their related heat dissipation equipment is eliminated. Chopper drives provide the highest torque speed levels, especially in frequent start-stop type applications, can extend the high speed slew range of the motors by a factor of 4-5 times and provide higher torque and speed in the start-stop mode.

The latter type drive is a constant current system.

A Central Processor Unit (CPU) is the control unit of a digital computer.

A Read Only Memory (ROM) is memory which is permanent or semi-permanent in nature and is not changeable by programming the computer and it is not lost when all electrical power goes off, i.e., it is non-volatile. It is used as fixed input to the computer. It's use is limited to being read by the computer. A PROM is programmable read only memory. It is semi-permanent in nature. It can be changed but the change is usually effected by special equipment when the memory is not installed in the using system.

A Random Access Memory (RAM) is memory which is changed during normal operation of the computer. The program can read this memory and can also change it. It is volatile so is lost if electrical power goes off.

APPENDIX B

INPUT CONTROL SYSTEM DESCRIPTION

General - This section provides discussion and diagrams to further define the system, including hardware and software, as it was finally developed.

Computer - The CPU diagram is shown in Figure B-1. The CPU plus memory, both ROM and RAM, make up the Altair 8800 computer. As will be shown in the software section, about 1000, 8 bit words of memory were required. The instruction set is shown in Figure B-2.

Icon 601-T Driver - This unit accepts a low power pulse stream on either of two input lines and transforms it into a sequential energization at high power of 5 output lines. If the outputs are numbered 1, 2, 3, 4 and 5 the pulse stream on one input line will cause the following pattern: 1,2 - 1,2,3 - 2,3 - 2,3,4 - 3,4 - 3,4,5 - 4,5 - 4,5, 1 - 5, 1 - 5,1,2 - and repeat. A pulse stream on the other input line will cause the following pattern: 1,2 - 5,1,2 - 5,1 - 4,5,1 - 4,5 - 3,4,5 - 3,4 - 2,3,4 - 2,3 - 1,2,3 - and repeat. This unit includes a 28-volt power supply. In an aircraft system the ship's 28-volt supply would be used. The circuit diagram for this unit is shown in Figure B-3. As noted in the basic report, this unit was modified to accept power from a current limiting type power supply which to a degree made the Icon 601-T driver perform like a chopper drive as defined in Appendix A. This modification allowed the external power supply to feed into the 5 power resistors in the EPM motor circuit while retaining the 12-volt internal supply for the 601-T logic circuits.

Encoders - The two encoders used were Model 25H-8NA made by Sequential. They are optical, absolute, and eight bit binary. They were connected as shown in Figure B-4.

Interface Board - The interface board schematic is shown in Figure B-5. This circuit performs the following functions:

- o digitizes the pressure transducer signals so the computer can read a binary number that represents pressure
- o converts two 8-bit binary words (i.e., 16 bits) output from the computer to direction and speed signals compatible with the Icon driver. This function is discussed further under the heading, Speed and Direction Control, which follows next in the text.
- o multiplexes the computer input and output lines to facilitate reading the two encoders and the pressure signal and outputting the speed, direction and shut-off valve signals.
- o Latches signals as necessary.

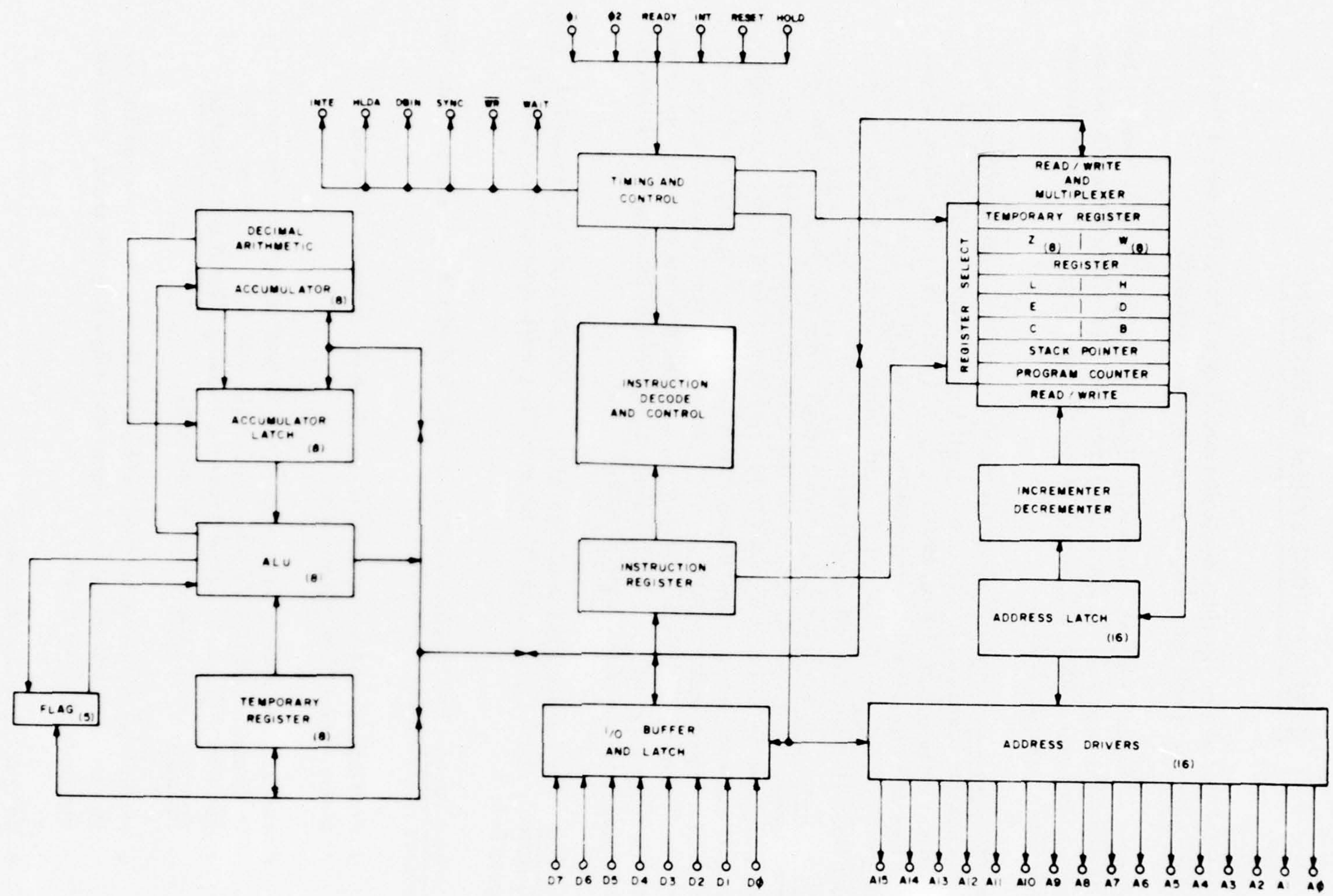


Figure B-1. CPU Diagram

Mnemonic	Description	Clock * Cycles
MOV $_{r1, r2}$	Move register to register	5
MOV M, r	Move register to memory	7
MOV r, M	Move memory to register	7
HLT	Halt	7
MVI r	Move immediate register	7
MVI M	Move immediate memory	10
INR r	Increment register	5
DCR r	Decrement register	5
INR M	Increment memory	10
DCR M	Decrement memory	10
ADD r	Add register to A	4
ADC r	Add register to A with carry	4
SUB r	Subtract register from A	4
SBB r	Subtract register from A with borrow	4
ANA r	And register with A	4
XRA r	Exclusive Or register with A	4
ORA r	Or register with A	4
CMP r	Compare register with A	4
ADD M	Add memory to A	7
ADC M	Add memory to A with carry	7
SUB M	Subtract memory from A	7
SBB M	Subtract memory from A with borrow	7
ANA M	And memory with A	7
XRA M	Exclusive Or memory with A	7
ORA M	Or memory with A	7
CMP M	Compare memory with A	7
ADI	Add immediate to A	7
ACI	Add immediate to A with carry	7
SUI	Subtract immediate from A	7
SBI	Subtract immediate from A with borrow	7
ANI	And immediate with A	7
XRI	Exclusive Or immediate with A	7
ORI	Or immediate with A	7
CPI	Compare immediate with A	7
RLC	Rotate A left	4
RRC	Rotate A right	4
RAL	Rotate A left through carry	4
RAR	Rotate A right through carry	4
JMP	Jump unconditional	10
JC	Jump on carry	10
JNC	Jump on no carry	10
JZ	Jump on zero	10
JNZ	Jump on no zero	10
JP	Jump on positive	10
JM	Jump on minus	10
JPE	Jump on parity even	10
JPO	Jump on parity odd	10
CALL	Call unconditional	17
CC	Call on carry	11/17
CNC	Call on no carry	11/17
CZ	Call on zero	11/17
CNZ	Call on no zero	11/17
CP	Call on positive	11/17
CM	Call on minus	11/17
CPE	Call on parity even	11/17
CPO	Call on parity odd	11/17
RET	Return	10
RC	Return on carry	5/11
RNC	Return on no carry	5/11

Mnemonic	Description	Clock * Cycles
RZ	Return on zero	5/11
RNZ	Return on no zero	5/11
RP	Return on positive	5/11
RM	Return on minus	5/11
RPE	Return on parity even	5/11
RPO	Return on parity odd	5/11
RST	Restart	11
IN	Input	10
OUT	Output	10
LXI B	Load immediate register Pair B & C	10
LXI D	Load immediate register Pair D & E	10
LXI H	Load immediate register Pair H & L	10
LXI SP	Load immediate stack pointer	10
PUSH B	Push register Pair B & C on stack	11
PUSH D	Push register Pair D & E on stack	11
PUSH H	Push register Pair H & L on stack	11
PUSH PSW	Push A and Flags on stack	11
POP B	Pop register pair B & C off stack	10
POP D	Pop register pair D & E off stack	10
POP H	Pop register pair H & L off stack	10
POP PSW	Pop A and Flags off stack	10
STA	Store A direct	13
LDA	Load A direct	13
XCHG	Exchange D & E, H & L Registers	4
XTHL	Exchange top of stack, H & L	18
SPHL	H & L to stack pointer	5
PCHL	H & L to program counter	5
DAD B	Add B & C to H & L	10
DAD D	Add D & E to H & L	10
DAD H	Add H & L to H & L	10
DAD SP	Add stack pointer to H & L	10
STAX B	Store A indirect	7
STAX D	Store A indirect	7
LDAX B	Load A indirect	7
LDAX D	Load A indirect	7
INX B	Increment B & C registers	5
INX D	Increment D & E registers	5
INX H	Increment H & L registers	5
INX SP	Increment stack pointer	5
DCX B	Decrement B & C	5
DCX D	Decrement D & E	5
DCX H	Decrement H & L	5
DCX SP	Decrement stack pointer	5
CMA	Compliment A	4
STC	Set carry	4
CMC	Compliment carry	4
DAA	Decimal adjust A	4
SHLD	Store H & L direct	16
LHLD	Load H & L direct	16
EI	Enable Interrupts	4
DI	Disable interrupt	4
NOP	No operation	4

*Two possible cycle times indicate that instruction cycles are dependent on condition flags. 4 Clock Cycles = 2μs

Figure B-2. Microcomputer Instruction Set

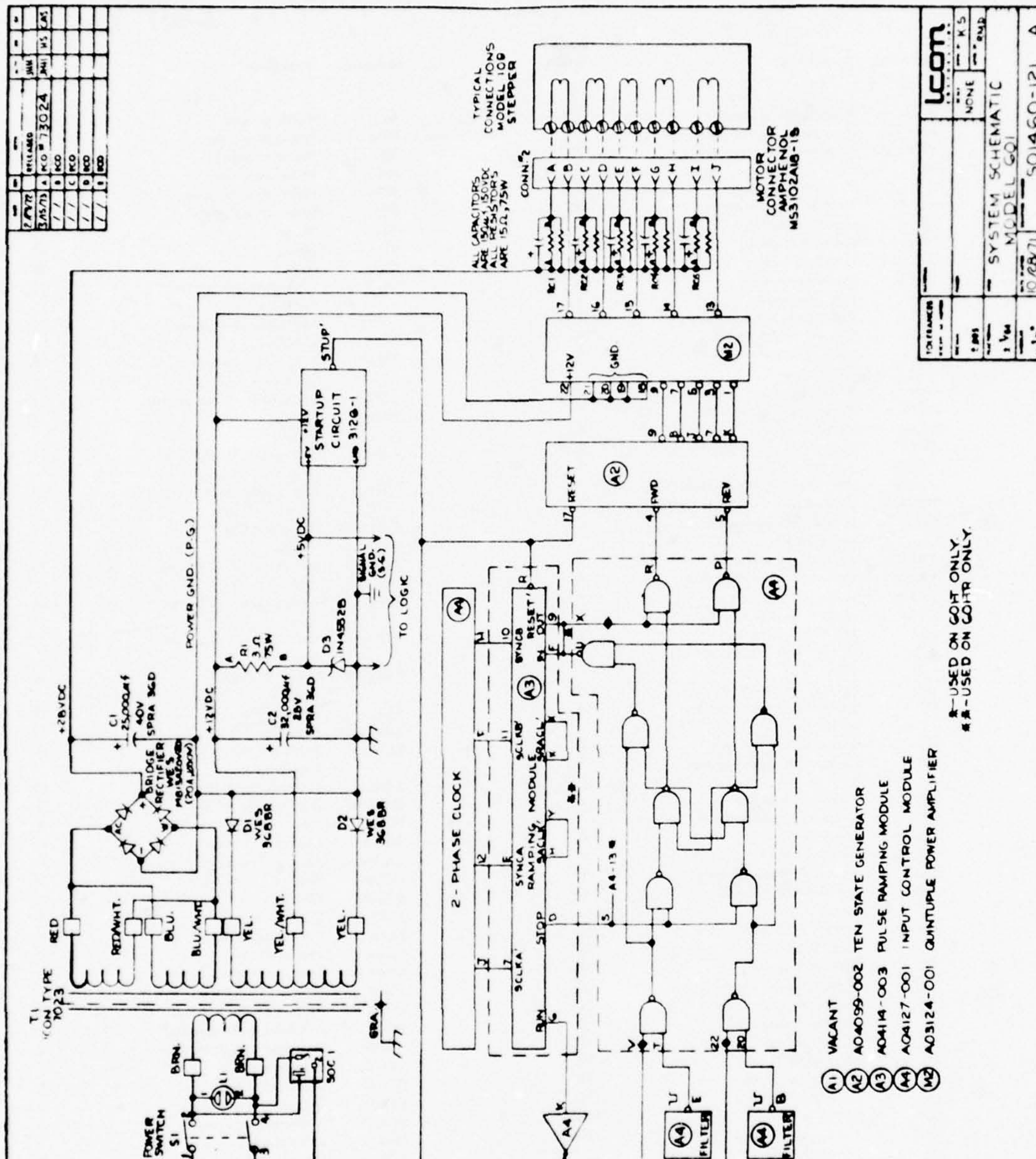


Figure B-3. ICON 601-T Driver

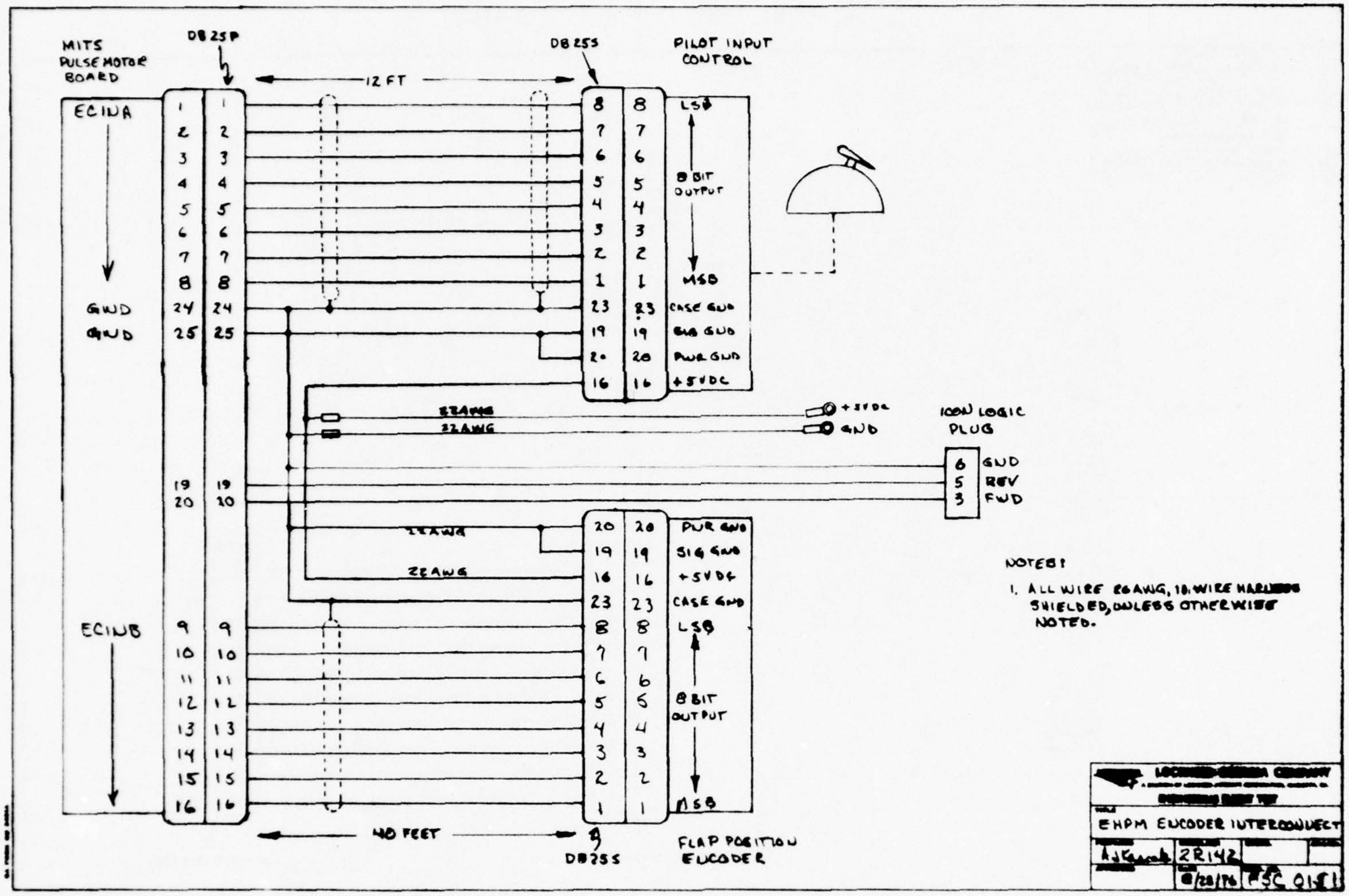


Figure B-4. Wiring Diagram (Sheet 1)

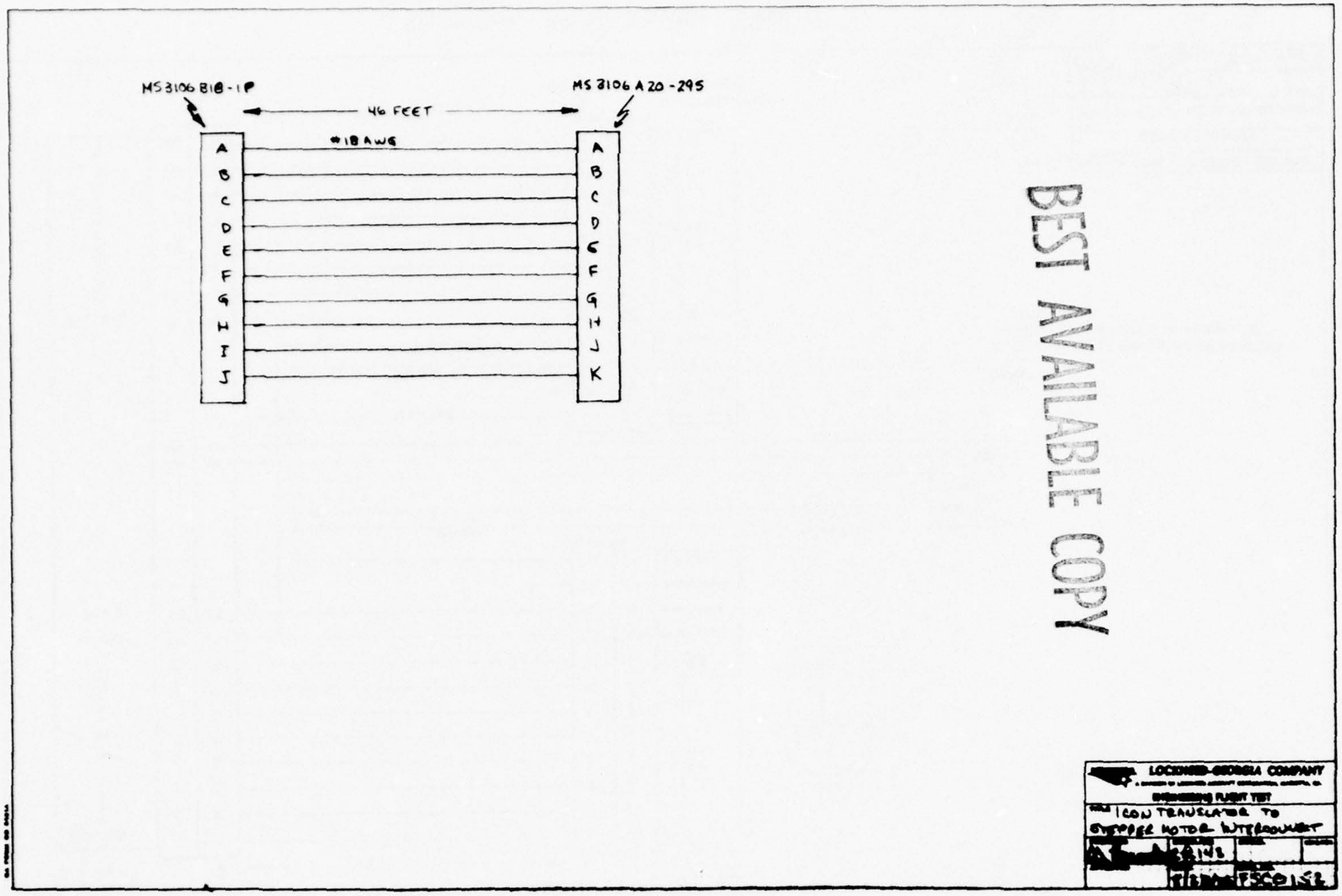


Figure B-4. Wiring Diagram (Sheet 2)

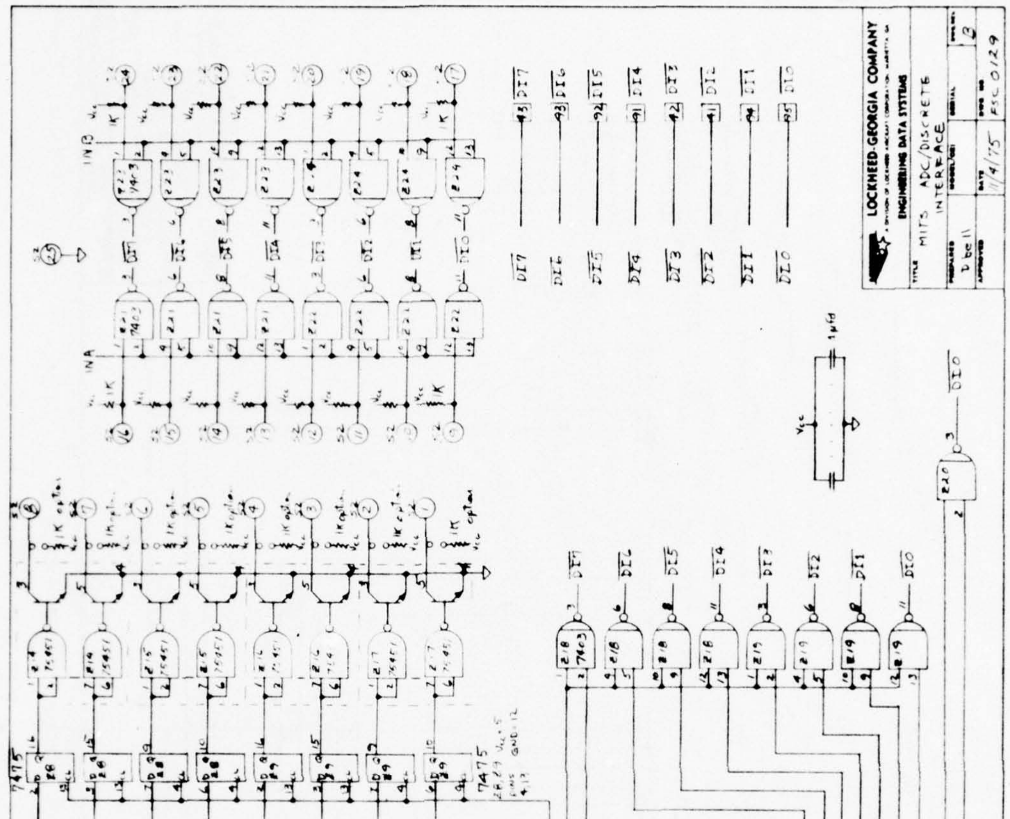


Figure B-5. Schematic Diagram of Interface Board (Sheet 1)

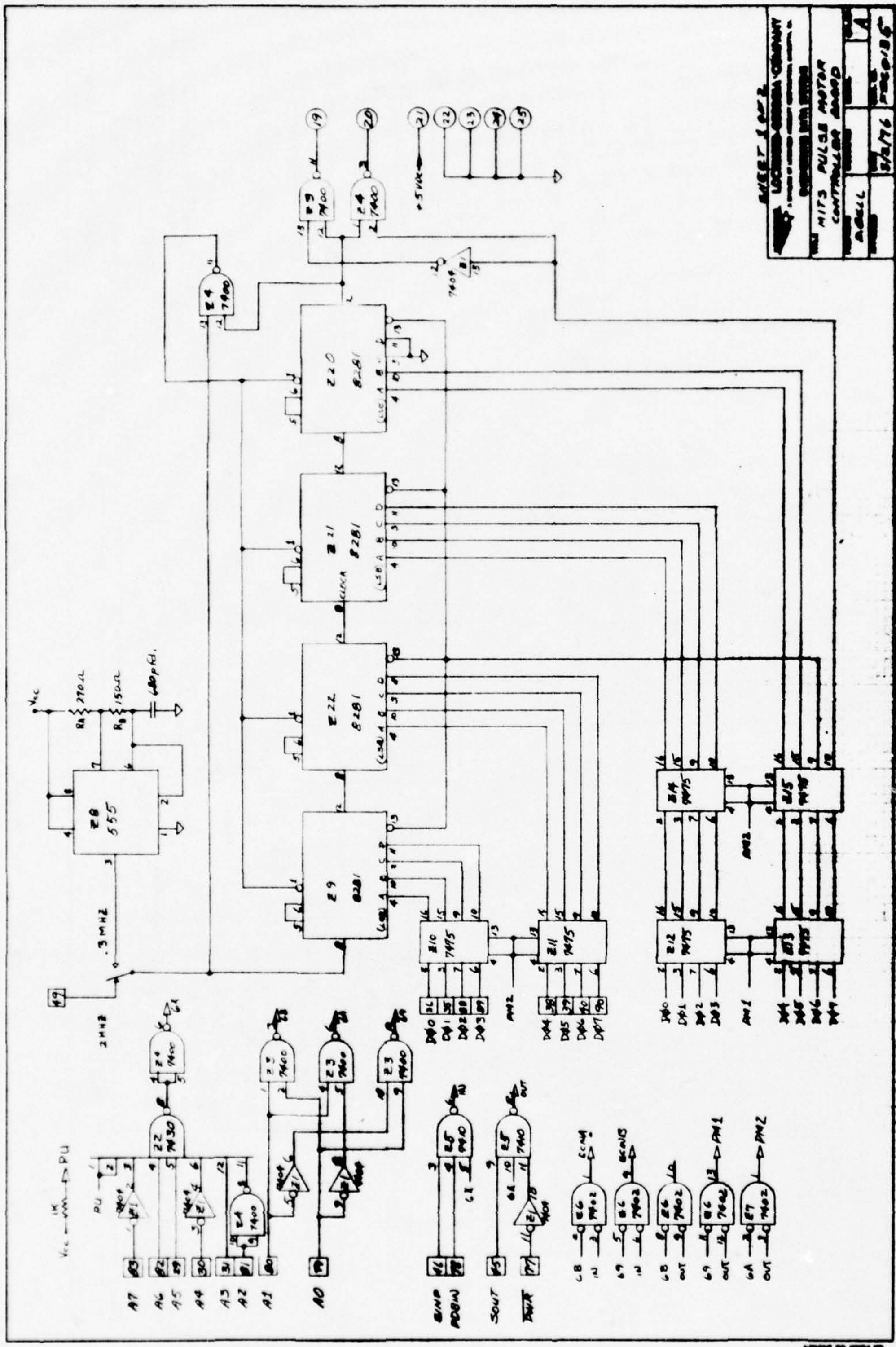


Figure B-5. Schematic Diagram of Interface Board (Sheet 2)

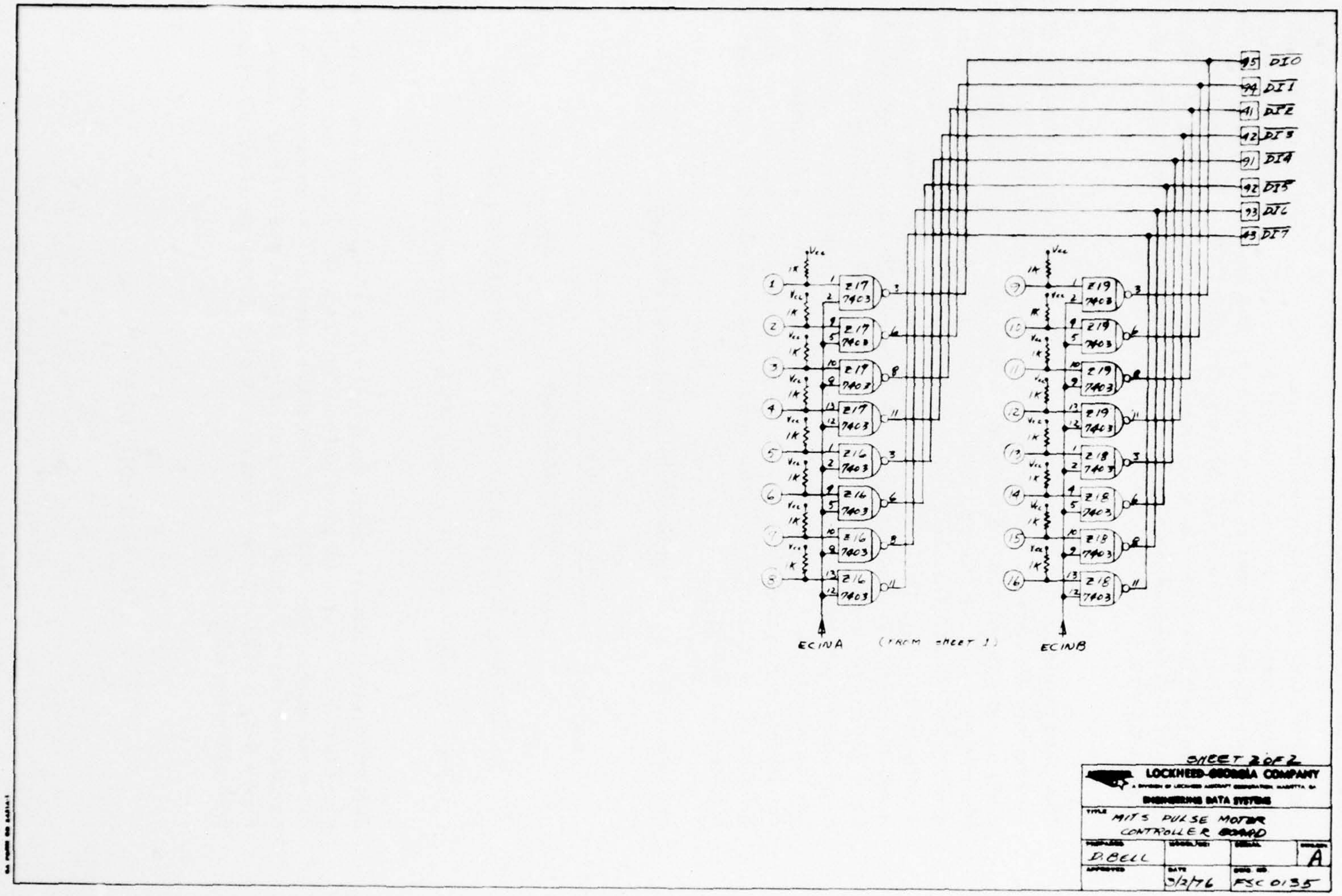


Figure B-5. Schematic Diagram of Interface Board (Sheet 3)

Speed and Directional Control - The function of the speed control circuit is to control the frequency of a pulse train. The computer works with binary numbers and uses a binary signal to represent speed. The speed control circuit transforms the binary number into a frequency or pulse rate which is related to the value of the binary number. Direction is controlled by steering this frequency onto one of two lines. One line causes CW rotation, the other CCW. These two lines are inputs to the 601-T driver which converts the low power pulse train to a sequential pattern of high power electrification of the 5 output lines of the driver, i.e., the coils of the EPM.

The speed control circuit in essence consists of a 14-bit binary counter, a clock feeding into the counter, a preset system which will preset the bits of the counter using a binary signal from the computer, and a logic system to determine when all bits of the counter are the same and deliver a pulse when they are the same. It operates as follows: The clock, which is operated at 2×10^6 pps, causes the counter to rip, i.e., the counter counts the clock pulses. When all bits are the same, the output line is energized causing one pulse in the pulse train that is the input to the Icon driver. The time between the pulses so generated depends on how many counts that are required to change the counter from the state where all bits are the same to the next state where all bits are the same. This number of counts can be selected by presetting various bits of the counter; i.e., the time required for the counter to count a full count is controlled and each time a full count occurs a pulse is generated.

The relationship is defined by the following speed equation:

$$\text{Speed (pps)} = \frac{\text{Clock rate}}{16{,}383 - \text{Preset number}}$$

16,383 = Base 10 value of full count in a 14-bit binary counter

Preset Number = Base 10 value of the binary preset number.

The binary preset number used in the program is a 16-bit number and the two most significant bits are used for direction and "go" control. The 14 least significant bits make up the preset number for the above equation. As an example, the hexadecimal number which is stored at addresses 016A and 016B of memory, Figure B-6, is FEB2 and constitutes one complete binary speed and directional control number which is

F	E	B	2
1111	1110	1011	0010

```
D,0,420

0000 : C3 50 00 00 00 00 00 00
0008 : 00 00 00 00 00 00 00 00
0010 : 00 00 00 00 00 00 00 00
0018 : 00 00 00 00 00 00 00 00
0020 : 00 00 00 00 00 00 00 00
0028 : 00 00 00 00 00 00 00 00
0030 : 00 00 00 00 00 00 00 00
0038 : 00 00 00 00 00 00 00 00
0040 : 31 FF 0F 3E F0 D3 FE FB
0048 : C3 80 02 00 AE C9 10 00
0050 : F5 3E D8 D3 FE 3A 4E 00
0058 : 3D 32 4E 00 C2 64 00 3E
0060 : 10 32 4E 00 F1 FB C9 00
0068 : 00 00 00 00 00 00 00 00
0070 : DB 70 E6 01 CA 70 00 D3
0078 : 70 DB 70 E6 01 CA 79 00
0080 : DB 71 2F 32 53 02 C9 00
0088 : 00 00 00 00 00 00 00 00
0090 : 3E 00 D3 69 D3 6A 3E C9
0098 : 32 4C 00 3A 4E 00 FE 02
00A0 : C2 9B 00 3D 32 4E 00 3A
00A8 : 4C 00 3D 32 4C 00 FE 00
00B0 : C2 9B 00 C3 80 02 00 00
00B8 : 00 00 00 00 00 00 00 00
00C0 : F3 3E 00 D3 69 D3 6A C3
00C8 : 00 10 00 00 00 00 00 00
00D0 : 2A 50 02 7C FE 05 C2 A0
00D8 : 02 7D FE 05 C2 A0 02 2A
00E0 : 59 02 7C FE A6 C2 A0 02
00E8 : 7D FE 01 C2 A0 02 3E 00
00F0 : D3 69 D3 6A 3E 80 D3 6F
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00F8 : C3 A0 02 00 00 00 00 00
0100 : FF 37 FF 36 FF 35 FF 34
0108 : FF 33 FF 32 FF 31 FF 30
0110 : FF 2F FF 2E FF 2D FF 2C
0118 : FF 2B FF 2A FF 29 FF 28
0120 : FF 27 FF 26 FF 25 FF 24
0128 : FF 23 FF 22 FF 21 FF 20
0130 : FF 1F FF 1E FF 1C FF 1B
0138 : FF 19 FF 17 FF 15 FF 13
0140 : FF 11 FF 0E FF 0C FF 09
0148 : FF 07 FF 04 FF 01 FE FE
0150 : FE FA FE F7 FE F4 FE F0
0158 : FE EB FE E6 FE E1 FE DB
0160 : FE D4 FE CC FE C8 FE BE
0168 : FE B9 FE B2 FE A7 FE 9E
0170 : FE 93 FE 8B FE 7D FE 71
0178 : FE 62 FE 52 FE 44 FE 34
0180 : FE 21 FE 0C FD F1 FD D6
0188 : FD C0 FD A1 FD 7B FD 50
0190 : FD 22 FC EC FC AC FC 66
0198 : FC 0D FB A1 FB 18 FA 68
01A0 : F9 7C F8 2F F5 94 00 00
01A8 : 75 94 78 2F 79 7C 7A 68
01B0 : 7B 18 7B A1 7C 0D 7C 66
01B8 : 7C AC 7C EC 7D 22 7D 50
01C0 : 7D 7B 7D A1 7D C0 7D D6
01C8 : 7D F1 7E 0C 7E 21 7E 34
01D0 : 7E 44 7E 52 7E 62 7E 71
01D8 : 7E 7D 7E 8B 7E 93 7E 9E
01E0 : 7E A7 7E B2 7E B9 7E BE
01E8 : 7E C8 7E CC 7E D4 7E DB
01F0 : 7E E1 7E E6 7E EB 7E F0
01F8 : 7E F4 7E F7 7E FA 7E FE
0200 : 7F 01 7F 04 7F 07 7F 09
0208 : 7F 0C 7F 0E 7F 11 7F 13
```

Figure B-6. Program Printout (Sheet 1)

```
0210 : 7F 15 7F 17 7F 19 7F 1B
0218 : 7F 1C 7F 1E 7F 1F 7F 20
0220 : 7F 21 7F 22 7F 23 7F 24
0228 : 7F 25 7F 26 7F 27 7F 28
0230 : 7F 29 7F 2A 7F 2B 7F 2C
0238 : 7F 2D 7F 2E 7F 2F 7F 30
0240 : 7F 31 7F 32 7F 33 7F 34
0248 : 7F 35 7F 36 7F 37 00 00
0250 : E3 E3 E3 BA 00 00 00 A6
0258 : 01 A6 01 4D 60 BA 00 00
0260 : 3A 5D 02 47 3A 53 02 4F
0268 : 32 5D 02 90 E6 0F 07 00
0270 : 00 00 47 79 90 47 3A 5C
0278 : 02 C9 00 00 00 00 00 00
0280 : 3E 00 D3 69 D3 6A 32 54
0288 : 02 32 55 02 3A 51 02 32
0290 : 52 02 2A 57 02 22 59 02
0298 : 3E C9 32 4D 00 00 00 00
02A0 : 3A 4E 00 FE 02 C2 A0 02
02A8 : 3D 32 4E 00 3A 4D 00 3D
02B0 : FE 00 C2 B7 02 3E 65 32
02B8 : 4D 00 FE 01 C2 D0 02 3A
02C0 : 52 02 47 3A 51 02 32 52
02C8 : 02 B8 CA D0 03 00 00 00
02D0 : DB 69 2F C6 03 32 51 02
02D8 : 47 2A 59 02 29 29 29 29
02E0 : 3E 1A 94 00 00 00 4F 78
02E8 : 91 47 DB 6B 32 50 02 90
02F0 : CA B0 03 DA 30 03 C3 70
02F8 : 03 00 00 00 00 00 00 00
0300 : 00 00 00 00 00 00 00 00
0308 : 2A 59 02 EB 13 13 1A D3
0310 : 69 32 54 02 13 1A D3 6A
0318 : 32 55 02 1B EB 22 59 02
0320 : C3 A0 02 2A 59 02 EB 1B
0328 : 1B C3 0E 03 00 00 00 00
0330 : CD 70 00 47 3A 5B 02 B8
0338 : D2 90 00 CD 60 02 B8 DA
0340 : 5A 03 2A 59 02 3A A2 01
0348 : BE C2 08 03 23 3A A3 01
0350 : BE C2 08 03 C3 A0 02 00
0358 : 00 00 2A 59 02 3A 22 01
0360 : BE C2 23 03 23 3A 23 01
0368 : BE 2B C2 23 03 C3 A0 02
0370 : CD 70 00 47 3A 5B 02 B8
0378 : D2 90 00 CD 60 02 B8 DA
0380 : 97 03 2A 59 02 3A A8 01
0388 : BE C2 23 03 23 3A A9 01
0390 : BE C2 23 03 C3 A0 02 2A
0398 : 59 02 3A 4C 02 BE C2 08
03A0 : 03 23 3A 4D 02 BE 2B C2
03A8 : 08 03 C3 A0 02 00 41 00
03B0 : 3A 54 02 FE 00 C2 C0 03
03B8 : 3A 55 02 FE 00 CA 80 02
03C0 : 3A 54 02 07 DA 08 03 C3
03C8 : 23 03 00 00 00 00 00 00
03D0 : 3E 00 D3 69 D3 6A 3E 20
03D8 : 32 4B 00 3A 4E 00 FE 02
03E0 : C2 DB 03 3D 32 4E 00 3A
03E8 : 4B 00 3D 32 4B 00 FE 00
03F0 : C2 DB 03 C3 80 02 00 00
03F8 : 00 00 00 00 00 00 00 00
0400 : 31 FF 00 CD 70 00 DB 69
0408 : 2F C6 04 32 51 02 DB 6B
0410 : 32 50 02 C3 00 10 41 00
0418 : 41 00 41 00 41 00 41 00
0420 : 41
DØNE
```

Figure B-6. Program Printout (Sheet 2)

Without the two most significant bits, the preset number has the base 10 value as follows:

$$1\quad 1\quad \overbrace{1\quad 1\quad 1\quad 0}^{E}\quad \overbrace{1\quad 0\quad 0\quad 1}^{B}\quad \overbrace{0\quad 0\quad 1\quad 0}^{2}$$

$$\begin{array}{r} 512 \\ 1024 \\ 2048 \\ 4096 \\ 8192 \\ 128 \\ 16 \\ \underline{2} \\ 16,050 \end{array}$$

which when substituted in the speed equation gives a speed of

$$\text{Speed} = \frac{2 \times 10^6}{16,383 - 16,050} = \frac{2 \times 10^6}{333} = 6006 \text{ pps}$$

Note that the adjacent binary numbers give:

$$\frac{2 \times 10^6}{332} = 6024 \text{ pps}$$

$$\frac{2 \times 10^6}{334} = 5988 \text{ pps}$$

also:

$$\frac{2 \times 10^6}{200} = 10,000 \text{ pps} = \text{Top Speed}$$

$$\frac{2 \times 10^6}{201} = 9,950 \text{ pps} = \text{Next to Top Speed}$$

so that 50 pps step change is the finest available at the upper speed whereas 6024 - 6006 = 18 pps step change is available at 6006 pps.

In summary, FE B2 says go CW at a speed of 6006 pps. If the number were 7E B2, which is in memory locations 01E2 and 01E3, the speed would be the same but the direction would be reversed. That is, the circled values in Figure B-6 are values stored at the indicated addresses in memory and when the program says output the value in one of these addresses the value stored (1111 1110 1011 0010 or 0111 1110 1011 0010) is delivered by the CPU to the speed control circuit which is located on the interface board.

Comparing FE B2 with 7E B2

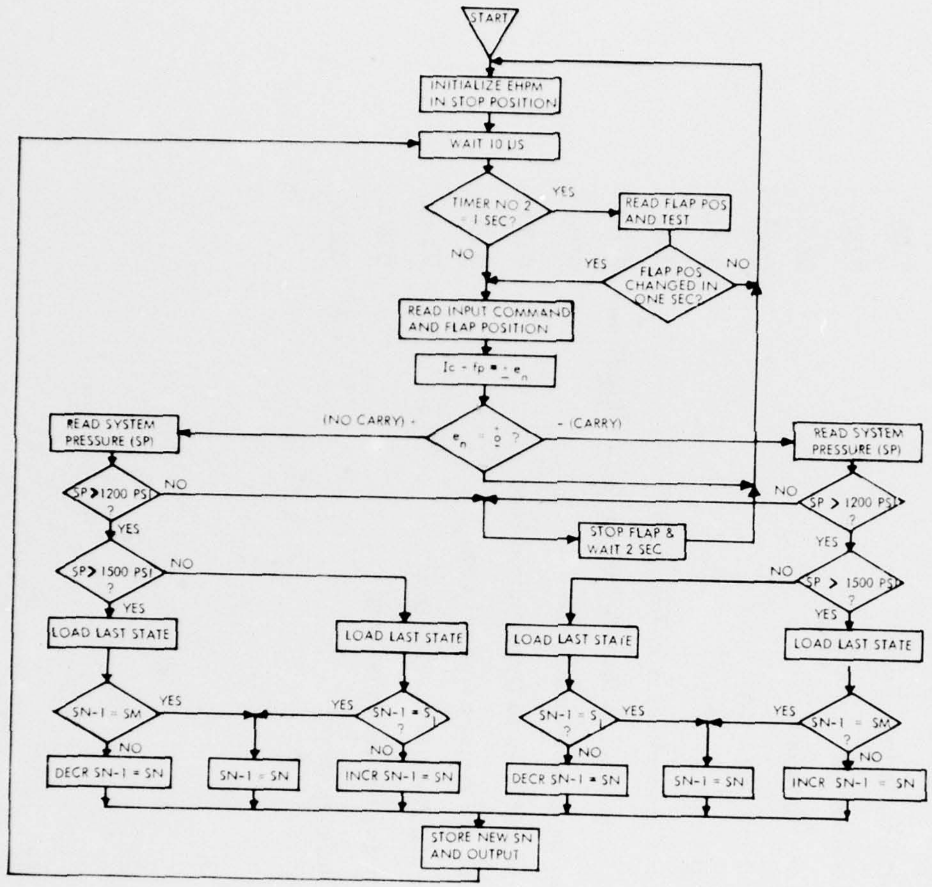
F = 1111

7 = 0111

The left digit is for directional control and the adjacent bit is "go". If it is 0, the unit will stop, i.e., either 1011 or 0011 will result in stop.

Software - Software is the list of instructions which tells the computer what to do. To prepare software it is a usual practice to first prepare a flow chart which pictorically defines the logic problem. The flow chart for the flap drive system is shown in Figure B-7. The computer is programmed using assembly language. A standard set of instructions called the instruction set provides the codes for programming. The Altair 8800 set was shown previously in Figure B-2. Binary numbers are difficult to work with so the computer is arranged to accept hexadecimal numbers. Using the flow chart and the instruction set the programmer develops an instruction arrangement which will cause the computer to handle the data in accordance with the logic of the flow chart. The program for the Flap Drive System is shown in Figure B-8. The computer reads this program and stores in memory the instructions it needs to accomplish the instructions.

For the Flap Drive System the information stored in memory was shown previously in Figure B-6. The computer stores the data in binary form but when it is instructed to display the data stored it transforms the binary data to hexadecimal form as shown in Figure B-6.



COMPUTER MEMORY STATE TABLE

-SM
-SM - 1
.
.
-SN
-SN-1
-S3
-S2
-S1
0
+S1
+S2
+S3
.
.
+ SN-1
+ SN
.
.
+ SM-1
+ SM

↑ LOWER MEMORY LOCATION

HIGHER MEMORY LOCATION ↓

S_M EXTREME STATE

S_N PRESENT STATE

SN-1 LAST STATE

Ic INPUT COMMAND

fp FLAP POSITION

$\pm e_n$ ERROR (PRESENT STATE)

Figure B-7 Iron Bird Flap Control Flow Chart

Index to Flap Drive System Programs of Figure B-8

	Address
Real Time Clock Trap	0000
Initialization of RTC	0040
Clock Counter Subroutine	0050
ADC Input Subroutine	0070
Low Pressure Time-out Loop	0090
Solenoid Valve Control Loop	00D0
Acceleration Table (non linear)	0100
Memory Location for Variables	0250
Pressure Loop Time-out	0260
Startup and Initialization	0280
Clock Wait Cycle	02A0
Clock Wait Cycle and Flap in Transit Test	02B7
Calculate Error Between Input & Position & Add Lead Term	02D0
Zero Error and Deceleration Loop	0308
Minus Error and Pressure Deceleration Loop	0330
Minus Error and Pressure Control Loop	035D
Plus Error and Pressure Control Loop	0370
Zero Error and Deceleration Loop	03B0
Restart Time-out Loop	03D0

```
FLAP DRIVE SYSTEM PROGRAM

REAL TIME CLOCK TRAP
0000  LEVEL 0:   JMP        C3
0001             COUNTER    50
0002             COUNTER    00
0003             NOP        00
0004             NOP        00
0005             NOP        00
0006             NOP        00
0007             NOP        00

INITIALIZATION OF RTC
0040             LXISP      31   INITIALIZE STACK
0041                        FF
0042                        OF
0043             MVIA       3E   SET INITIALIZATION BITS
0044                        FO
0045             OUT        D3   OUTPUT TO VI BOARD
0046             VI         FE
0047             EI         FB   ENABLE INTERRUPT
0048             JMP        C3   USER PROGRAM
0049             START      80
004A                        02
004B             DATA       00   DC COUNTER NO. 4
004C             DATA       00   DC COUNTER NO. 3
004D             DATA       00   DC COUNTER NO. 2
004E             DATA       XX   DC COUNTER NO. 1
004F             DATA       00   DC COUNTER NO. 5

CLOCK COUNTER SUBROUTINE
0050  COUNTER:   PUSH PSW   F5   SAVE A & FLAGS
0051             MVIA       3E   RESET INTERRUPT LEVEL
0052                        D8
0053             OUT        D3   OUTPUT CLOCK
0054             VI         FE
0055             LDA        3A   FETCH COUNTER
0056                        4E
0057                        00
0058             DCRA       3D   DECREMENT COUNTER
0059             STA        32   SAVE COUNTER
005A                        4E
005B                        00
005C             JNZ        C2
005D             OUT        64
005E                        00
005F             MVIA       3E   RESET COUNTER
0060             DATA       XX   (MSEC/COUNT +1)
0061             STA        32   SAVE NEW COUNT
0062                        4F
0063                        00
0064  OUT:       POP PSW    F1   RESTORE A & FLAGS
0065             EI         FB   ENABLE INTERRUPT
0066             RET        C9
0067             NOP        00

ADC INPUT SUBROUTINE
0070  ADC:       IN         DB   PORT IN ADC
0071             70         70
0072             ANI        E6   MASK FOR STATUS
0073                        01
0074             JZ         CA   WAIT FOR READY
0075             ADC        70
0076                        00
0077             OUT        D3   START CONVERSION
0078                        70
0079  CONV:      IN         DB   PORT IN ADC AGAIN
007A                        70
007B             ANI        E6   MASK FOR STATUS
007C                        01
007D             JZ         CA   WAIT FOR CONVERSION
007E             CONV       79
007F                        00
0080             IN         DB   PORT  IN ADC
0081                        71
0082             CMA        2F   INVERT INPUT
0083             STA        32   SAVE ADC INPUT
0084             ADC        53   IN "PRESS"
0085                        02
0086             RET        C9

LOW PRESSURE TIME OUT LOOP
0090  DELAY:                3E   LOAD   A   WITH ZERO
0091                        00
0092             OUT        D3   OUTPUT PMI  STOP EHPM
0093             FYI        69
0094             OUT        D3   OUTPUT PM2  STOP EHPM
0095             PN2        6A
0096             OUT        D3   SHUT OFF VALVE
0097                        6F
0098                        3E   INITIALIZE COUNTER NO. 3
0099                        C9   2 SEC.
009A             STA        32   SAVE COUNT IN CTR2
009B             CTR2       4C
009C                        00
009D  RETRY:     LDA        3A   FETCH CTR1
009E             CTR1       4E
009F                        00
00A0             CPI        FE   TEST COUNTER
00A1                        02   CTR1 = 2
00A2             JNZ        C2   NO? LOOP TO  RETRY
00A3             RETRY      9D
00A4                        00
00A5             DCRA       3D   FORCE CTR DOWN & COUNT
00A6             STA        32   SAVE NEW COUNT
00A7                        4E
00A8                        00
00A9             LDA        3A   FETCH COUNTER NO. 3
00AA                        4C
00AB                        00
00AC             DCRA       3D   DECREMENT CTR NO. 3
00AD             STA        32   SAVE NEW COUNT
00AE                        4C
00AF                        00
00B0             CDI        FE   TEST CTR NO. 3
00B1                        00   CTR B = 0?
00B2             JNZ        C2   NO LOOP TO RETRY
00B3             RETRY      9D
00B4                        00
00B5             JMP        C3
00B6             START      EO
00B7                        02

SOLENOID VALVE CONTROL LOOP
0D0   VALVE:     LHLD       2A   FETCH INCMD & FLPOSI
0D1              INCMP      50
0D2                         02
0D3              MOVAH      7C   MOVE INCMD TO A
0D4              CPI        FE   IS INCMD = 05?
0D5                         05
0D6              JNZ        C2   NO? GO TO OPEN
0D7              WAIT       AO
0D8                         02
0D9              MOVAL      7D   MOV FLPOS 1 to A
0DA              CPI        FE   IS FLPOS = 05?
0DB                         05
0DC              JNZ        C2   NO? GO TO OPEN
0DD              WAIT       AO
0DE                         02
0DF              LHLD       2A   FETCH PTR
0E0              PTR        59
0E1                         02
0E2              MOVAH      7C   MOVE PTR HI ADDRESS TO A
0E3              CPI        FE   IS PTR HI ADDRESS = ZERO?
0E4                         A6
0E5              JNZ        C2   NO? GO TO OPEN
0E6              WAIT       AO
0E7                         02
0E8              MOVAL      7D   MOVE PTR LO ADDRESS TO A
0E9              CPI        FE   IS PTR LO ADDRESS = ZERO +1?
0EA                         01
0EB              JNZ        C2   NO? GO TO OPEN
0EC              WAIT       AO
0ED                         02
0EE              MVIA       3E   LOAD A REG WITH ZERO
0EF                         00   TO STOP PULSE MOTOR
0F0              OUT        D3   OUTPUT PM1
0F1                         69
0F2              OUT        D3   OUTPUT PM2
0F3                         6A
0F4              MVIA       3E   OK TO SHUT OFF VALVE
0F5                         00   SET OUTPUT BIT
0F6              OUT        D3   SHUT OFF VALVE
0F7                         6F
0F8              JMP        C3
0F9              WAIT       AO
0FA                         02
0FB              NOD        00
0FC              NOP        00
0FD              NOP        00
0FE              NOP        00
0FF              NOP        00
```

Figure B-8. C-5 Iron Bird Flap Drive Program (Sheet 1)

ACCELERATION TABLE (NON LINEAR)

Address	MNSSM	DATA	
0100		FF	10,000 PPS
0101		37	
0102		FF	9,950
0103		36	
0104		FF	9,900
0105		35	
0106		FF	9,850
0107		34	
0108		FF	9,800
0109		33	
010A		FF	9,750
010B		32	
010C		FF	9,700
010D		31	
010E		FF	9,650
010F		30	
0110		FF	9,600
0111		2F	
0112		FF	9,560
0113		2E	
0114		FF	9,530
0115		2D	
0116		FF	9,495
0117		2C	
0118		FF	9,440
0119		2B	
011A		FF	9,390
011B		2A	
011C		FF	9,345
011D		29	
011E		FF	9,315
011F		28	
0120		FF	9,270
0121		27	
0122		FF	9,200
0123		26	
0124		FF	9,150
0125		25	
0126		FF	9,115
0127		24	
0128		FF	9,085
0129		23	
012A		FF	9,055
012B		22	
012C		FF	9,025
012D		21	
012E		FF	8,970
012F		20	
0130		FF	8,935
0131		1F	
0132		FF	8,895
0133		1E	
0134		FF	8,820
0135		1C	
0136		FF	8,770
0137		1B	
0138		FF	8,690
0139		19	
013A		FF	8,630
013B		17	
013C		FF	8,530
013D		15	
013E		FF	8,480
013F		13	

0140		FF	8,400
0141		11	
0142		FF	8,310
0143		0E	
0144		FF	8,230
0145		0C	
0146		FF	8,140
0147		09	
0148		FF	8,070
0149		07	
014A		FF	7,975
014B		04	
014C		FF	7,880
014D		01	
014E		FF	7,780
014F		FE	
0150		FE	7,670
0151		FA	
0152		FE	7,590
0153		F7	
0154		FE	7,480
0155		F4	
0156		FE	7,370
0157		F0	
0158		FE	7,260
0159		EB	
015A		FE	7,110
015B		E6	
015C		FE	7,000
015D		E1	
015E		FE	6,860
015F		DB	
0160		FE	6,700
0161		D4	
0162		FE	6,560
0163		CC	
0164		FE	6,430
0165		C8	
0166		FE	6,290
0167		BE	
0168		FE	6,140
0169		B9	
016A		FE	6,000
016B		B2	
016C		FE	5,820
016D		A7	
016E		FE	5,670
016F		9E	
0170		FE	5,500
0171		93	
0172		FE	5,370
0173		8B	
0174		FE	5,185
0175		7D	
0176		FE	5,025
0177		71	
0178		FE	4,840
0179		62	
017A		FE	4,700
017B		52	
017C		FE	4,510
017D		44	
017E		FE	4,360
017F		34	
0180		FE	4,180

0181		21	
0182		FE	4,005
0183		0C	
0184		FD	3,800
0185		F1	
0186		FD	3,650
0187		D6	
0188		FD	3,480
0189		C0	
018A		FD	3,300
018B		A1	
018C		FD	3,105
018D		7B	
018E		FD	2,900
018F		50	
0190		FD	2,730
0191		22	
0192		FC	2,540
0193		EC	
0194		FC	2,350
0195		AC	
0196		FC	2,176
0197		66	
0198		FC	1,980
0199		0D	
019A		FB	1,788
019B		A1	
019C		FB	1,594
019D		18	
019C		FA	1,398
019F		68	
01A0		F9	1,200
01A1		7C	
01A2	MIN PLS	F8	1,000
01A3		2F	
01A4		F5	750
01A5		94	
01A6	ZZ	00	00
01A7	ZZ + 1	00	
01A8	MIN MNS	75	750
01A9		94	
01AA		78	1,000
01AB		2F	
01AC		79	1,200
01AD		7C	
01AE		7A	1,398
01AF		68	
01B0		7B	1,594
01B1		18	
01B2		7B	
01B3		A1	
01B4		7C	
01B5		0D	
01B6		7C	
01B7		66	
01B8		7C	
01B9		AC	
01BA		7C	2,540
01BB		EC	
01BC		7D	
01BD		22	
01BE		7D	
01BF		50	
01C0		70	
01C1		7B	

01C2		7D	
01C3		A1	
01C4		7D	
01C5		C0	
01C6		7D	
01C7		D6	
01C8		7D	
01C9		F1	
01CA		7E	
01CB		0C	
01CC		7E	
01CD		21	
01CE		7E	
01CF		34	
01D0		7E	
01D1		44	
01D2		7E	
01D3		52	
01D4		7E	
01D5		62	
01D6		7E	5,025
01D7		71	
01D8		7E	
01D9		7D	
01DA		7E	
01DB		8B	
01DC		7E	
01DD		93	
01DE		7E	
01DF		9E	
01E0		7E	
01E1		A7	
01E2		7E	
01E3		B2	
01E4		7E	
01E5		B9	
01E6		7E	
01E7		BE	
01E8		7E	
01E9		C8	
01EA		7E	
01EB		CC	
01EC		7E	
01ED		D4	
01EE		7E	
01EF		DB	
01F0		7E	
01F1		E1	
01F2		7E	
01F3		E6	
01F4		7E	
01F5		EB	
01F6		7E	
01F7		F0	
01F8		7E	7,480
01F9		F4	
01FA		7E	
01FB		F7	
01FC		7E	
01FD		FA	
01FE		7E	
01FF		FE	
0200		7F	
0201		01	
0202		7F	

Figure B-8. C-5 Iron Bird Flap Drive Program (Sheet 2)

```
0203    04
0204    7F
0205    07
0206    7F
0207    09
0208    7F
0209    00
020A    7F
020B    0E
020C    7F
020D    11
020E    7F    8, 480
020F    13
0210    7F
0211    15
0212    7F
0213    17
0214    7E
0215    19
0216    7F
0217    1B
0218    7F
0219    1C
021A    7F
021B    1E
021C    7F
021D    1F
021E    7F
021F    20
0220    7F    9, 025
0221    21
0222    7F
0223    22
0224    7F
0225    23
0226    9F    9, 150
0227    24
0228    7F
0229    25
022A    7F    9, 200
022B    26
022C    7F
022D    B7
022E    7F
022F    28
0230    7F
0231    29
0232    7F
0233    2A
0234    7F
0235    2B
0236    7F
0237    2G
0238    7F
0239    2D
023A    7F
023B    2E
023C    7F
023D    2F
023E    7F
023F    30
0240    7F
0241    31
```

```
0242                      7F
0243                      32
0244                      7F
0245                      33
0246                      7F
0247                      34
0248                      7F
0249                      35
024A                      7F
024B                      36
024C  PLSSM               7F
024D                      37
024E            NOP       00
024F            NOP       00
```

MEMORY LOCATION FOR VARIABLES

```
0250  INCMD     DATA      XX   HANDLE INPUT
0251  FLPOS     DATA      XX   FLAP POSITION
0252  FLPOS 2   DATA      XX
0253  ADC       DATA      XX
0254  LSTE1     DATA
0255  LSTE2     DATA
0256  ERROR     DATA
0257  ZRO       DATA      A6
0258  ZRO + 1   DATA      01
0259  PTR       DATA      XX
025A  PTR+1     DATA      XX
025B  PS1200              4D
025C  PS1500              60
025D  PS2000    DATA      80
025C  PS2       DATA      XX
```

PRESSURE LOOP TIMEOUT

```
260   DLTA 5    LDA       3A   FFTCH CTR 1
261             CTR 1     4E
262                       00
263             CPI       FE   TEST CTR 4
264                       02   CTR 1 = 02?
265             JNZ       C2   NO? LOOP TO DLTA 5
266             DLTA 5    60
267                       02
268             DCRA      3D   FORCE COUNTER DOWN 1 COUNT
269             STA       32   SAVE CTR-1
26A             CTR 1     4E
26B                       00
26C             LDA       3A   FETCH CTR 5
26D             CTR 5     4F
26E                       00
26F             CPI       FE   TEST CTR 5
270                       00   CTR 5 = 0?
271             JNZ       C2   NO? LOOP TO DLTA 5
272             DLTA 3    60
273                       02
274             RET       C9   END OF TIMEOUT
```

START UP & INITIALIZATION

```
280   START     MVIA      3E   LOAD A REG WITH ZERO
281                       00
282             OUT       D3   OUTPUT PM1
283             PM1       69
284             OUT       D3   OUTPUT PM2
285             PM2       6A
```

```
286             STA       32   PUT ZERO IN LSTE 1
287             LSTE 1    54
288                       02
289             STA       32   PUT ZERO IN LSTE 2
28A             LSTE 2    55
28B                       02
28C             LDA       3A   PUT ZERO IN FLPOS 1
28D             FLPOS 1   51
28E                       02
28F             STA       32   PUT ZERO IN FL POS2
290             FLPOS2    52
291                       02
292             LHLD      2A   FETCH INITIAL VALUE FOR
293             ZRO       57   POINTER
294                       02
295             SHLD      22   INITIALIZE POINTER
296             PTR       57
297                       02
298             MVIA      3E   INITIALIZE COUNTER NO. 2
299                       65
29A             STA       32
29B                       4D
29C                       00
29D             JMP       C3
29E             VALVE     DO
29F                       00
```

CLOCK WAIT CYCLE

```
2A0   WAIT      LDA       3A   FETCH COUNTER NO. 1
2A1             CTR 1     4E
2A2                       00
2A3             CPI       FE   TEST COUNTER
2A4                       02   COUNTER = 2?
2A5             JNZ       C2   NO? LOOP TO WAIT
2A6             WAIT      AO
2A7                       02
2A8             DCRA      3D   FORCE COUNTER DOWN 1 COUNT
2A9             STA       32   SAVE NEW COUNT
2AA                       4E
2AB                       00
2AC             LDA       3A   FETCH COUNTER NO. 2
2AD                       4D
2AE                       00
2AF             DCRA      3D   DECREMENT CTR BY 1 COUNT
2B0             CPI       FE   TEST COUNTER NO. 2
2B1                       00   CTR 2 = 0?
2B2             JNZ       C2   NO? JUMP TO Z NOT
2B3             Z NOT     B7
2B4                       02
2B5             MVIA      3E   RESET COUNTER NO. 2
2B6                       65
```

CLOCK WAIT CYCLE & FLAP IN TRANSIT TEST

```
2B7   Z NOT     STA       32   SAVE COUNTER
2B8                       4B
2B9                       00
2BA             CPI       FE   TEST COUNTER
2BB                       00   CTR NO. 2 = 0?
2BC             JNZ       C2   NO-GO TO INPUT
2BD             INPUT     DO
2BE                       02
2BF             LDA       3A   FETCH FLPOS 2
```

Figure B-8. C-5 Iron Bird Flap Drive Program (Sheet 3)

```
2C0                      52
2C1                      02
2C2          MOVBA       47   MOVE FLPOS 2 TO REG B
2C3          LDA         3A   FETCH FLPOS 1
2C4                      51
2C5                      02
2C6          STA         32   MAKE FLPOS 1 = FLPOS 2
2C7                      52
2C8                      02
2C9          CMPB        B8   IS FLPOS 2 = FLPOS 1
2CA          JZ          CA   YES? FLAP NOT MOVING
2CB          START       80
2CC                      02   NO? FLAP MOVING - CONTINUE
2CD          NOP         00
2CE          NOP         00
2CF          NOP         00
```

CALCULATE ERROR BETWEEN INPUT & POSITION & ADD LEAD TERM

```
2D0  INPUT:  IN          DB   INPUT FLAP POSITION
2D1          ECINB       69
2D2          CMA         2F   INVERT ENCODER OUTPUT
2D3          ADI         C6   COMPENSATE FOR ENCODER
2D4                      04   INSTALLATION ERROR
2D5          STA         32   SAVE IN FL POS 1
2D6          FLPOS 1     51
2D7                      02
2D8          MOVBA       47   FLPOS 1 to REG B
2D9          LHLD        2A   FETCH POINTER
2DA          PTR         59
2DB                      02
2DC          DADH        29   SHIFT LEFT 4 BITS
2DD          DADH        29   THRU H & L REGISTERS
2DE          DADH        29
2DF          DADH        29
2E0          MVIA        3E   PUT PTR ZERO IN A REG
2E1                      1A   OD 06
2E2          SUBH        94   (1A - PTR)/8 = LEAD
2E3          MOVCA       4F   LEAD TO C REG
2E4          MOV AB      78   FLPOS 1 TO A REG
2E5          SUB C       91   FLPOS - LEAD
2E6          MOV BA      47   (FLPOS - LEAD) TO B REG
2E7          IN          DB   INPUT FLAP HANDLE
2E8          ECINA       68
2E9          STA         32   SAVE HANDLE POS IN INCMD
2EA          INCMO       50
2EB                      02
2EC          SUB B       90   INCMD - (FLPOS - LEAD)
2ED          JZ          CA   START DECEL
2EE          EZRO        B0
2EF                      02
2F0          MVIA        3E   ERROR NET ZERO
2F1                      80   SET VALVE BIT
2F2          OUT         D3   OPEN VALVE
2F3                      6F
2F4          JC          DA   ERROR IS MINUS
2F5          MUUSER      30
2F6                      02
2F7          JMP         C3   ERROR IS PLUS
2F8          PLUS CIL    70
2F9                      03
```

ZERO ERROR & DECELERATION LOOP

```
308  INCR:   LHLD        2A   PUT TABLE ADDRESS IN H, L
309          PTR         59
30A          XCHG        EB   MOVE TABLE ADDRESS TO D, E
30C          INXD        13   ADVANCE 2 STEPS IN TABLE
30D          INXD        13
30E  OUTPUT: LDAXD       1A   PUT TABLE CONTENTS IN A REG
30F          OUT         D3   OUTPUT PM 1
310          PM1         69
311          STA         32   SAVE TABLE CONTENT IN LSTE 1
312          LSTE 1      54
313                      02
314          INXD        13   ADVANCE 1 STEP IN TABLE
315          LDAXD       1A   PUT TABLE CONTENTS IN A REG
316          OUT         D3   OUTPUT PM 2
317          PM 2        6A
318          STA         32   SAVE TABLE CONTENTS IN LSTE 2
319          LSTE 2      55
31A                      02
31B          DCXD        1B   BACK UP 1 STEP IN TABLE
31C          XCHG        EB   MOVE TABLE ADDRESS BACK TO H, L
31D          SHLD        22   SAVE TABLE ADDRESS IN PTR
31E          PTR         59
31F                      02
320          JMP         C3
321          WAIT        A0
322                      02
323  DECR:   LHLD        2A   PUT TABLE ADDRESS IN H, L
324          PTR         59
325                      02
326          XCHG        EB   MOVE TABLE ADDRESS TO D, E
327          DCXD        1B   BACK UP 2 STEPS IN TABLE
328          DCXD        1B
329          JMP         C3   LOOP TO OUTPUT
32A          OUTPUT      0E
32B                      03
32C          NOP         00
32D          NOP         00
32E          NOP         00
32F          NOP         00
```

MINUS ERROR & PRESSURE DECELERATION LOOP

```
330  MNUSER  CALL        CD   GET PS FROM ADC
331          ADC         70
332                      00
333          MOVBA       47   PUT PS IN B REG
334          LDA         3A   FETCH PS 1200
335          PS1200      5B   9E
336                      02
337          CMPB        B8   (B > A = SET CARRY)
338          JNC         D2   PS IS BELOW 1200 PSI
339          DELAY       90   STOP EVERYTHING FOR 2 SEC
33A                      00
33B          LDA         3A   FETCH PS 1500
33C          PS1500      5C
33D                      02
33E          CMPB        B8   PS ABOVE 1500 PSI?
33F          JIC         DA   YES   GO TO TEST HI
340          TST HI      5A
341                      03
342          LHLD        2A   NO PS IS BELOW 1500
343          PTR         59   FETCH PTR
344                      02
345          LDA         3A   FETCH MIN PLS
346                      A4
347                      01
348          CMPM        BE   FIRST WORD ZERO?
349          JNZ         C2   NO-GO TO INCR
34A          INCR        08
34B                      03
34C          INXH        23   INCREMENT PTR
34D          LDA         3A
34E          MIN PLS +1  A5
34F                      01
350          CMPM        BE   2ND WORD ZERO?
351          JNZ         C2   NO-GO TO INCR
352          INCR        08
B53                      03
354          JMP         C3   YES JMP TO WAIT
355          WAIT        A0
356                      02
357          NOP         00
358          NOP         00
359          NOP         00
35A  TST HI: LHLD        2A
35B          PTR         59
35C                      02
35D          LDA         3A   HIGHEST VALUE IN TABLE
35E          MNSSM       00
35F                      01
360          CMPM        BE   IS PTR AT HIGHEST VALUE?
361          JNZ         C2   NO?   GO TO DECR
362          DECR        23
363                      03
364          INXH        23   YES? FETCH NEXT WORD
365          LDA         3A   HIGHEST VALUE IN TABLE MINUS 1
366          MNSST       01
367                      01
368          CMPM        BE   IS PTR AT HI VALUE -1?
369          DCXH        2B   RESET PTR TO HI VALUE
36A          JNZ         C2   NO?   TO TO DECR
35B          DECR        23
36C                      03
36D          JMP         C3   YES?   GO TO WAIT
36E          WAIT        A0
36F                      02
```

PLUS ERROR & PRESSURE CONTROL LOOP

```
370  PLUSER: CALL        CD   GET PS FROM ADC
371          ADC         70
372                      00
373          MOVBA       47   PUT PS IN B REGISTER
374          LDA         3A   FETCH PS 1200
375          PS 1200     5B
376                      02
377          CMP B       B8   IS PS ABOVE OR BELOW 1200?
378          JNC         D2   BELOW - STOP EVERYTHING FOR 2 SEC.
379          DELAY       90
37A                      00
37B          LDA         3A   ABOVE - FETCH PS 1500
37C          PS 1500     5C
37D                      02
37E          CMPB        B8   IS PS ABOVE OR BELOW 1500?
```

Figure B-8. C-5 Iron Bird Flap Drive Program (Sheet 4)

```
37F                JC         DA   YES GO TO HOLD
380                TST LO     97
381                           03
382                LHLD       2A   NO PS IS BELOW 1500
383                PTR        59
384                           02
385                LDA        3A
386                           AC
387                           01
388                CMPM       BE   FIRST WORD ZERO?
389                JNC        C2   NO-GO TO DECR
38A                DECR       23
38B                           03
38C                INXH       23   INCREMENT PTR
38D                LDA        3A
38C                M.: MINS +1 AD
38F                           01
390                CMPM       BE   2ND WORD ZERO?
391                JNZ        C2   NO-GO TO DECR
392                DECK       23
393                           03
394                JMD        C3   YES - JMP TO WAIT
395                WAIT       AO
396                           02
397     TST LO:    LHLD       2A   FETCH POINTER
398                PTR        59
399                           02
39A                LDA        3A   LOWEST VALUE IN TABLE
39B                PLSSM      4C
39C                           02
39D                CMPM       BE   IS PTR = LOWEST VALUE?
39E                JNZ        C2   NO? GO TO INCR
39F                INCR       08
3A0                           03
3A1                INXH       23   YES? FETCH NEXT WORD
3A2                LDH        3A   LOWEST VALUE + 1
3A3                PLSSM      4D
3A4                           02
3A5                CHPM       BE   IS PTR = LOWEST VALUE + 1?
3A6                DCXH       2B   RESET PTR TO LO VALUE
3A7                JNC        C2   NO? GO TO INCR
3A8                INCR       08
3A9                           03
3AA                JMP        C3   YES -GO TO WAIT
3AB                WAIT       AO
3AC                           02
3AD
3AE
3AF

ZERO ERROR & DECELERATION LOOP

3B0     E ZRO:     LDA        3A   FETCH LAST TABLE ENTRY
3B1                LSTE1      54
3B2                           02
3B3                CPI        FE   IS ENTRY ZERO?
3B4                           00
3B5                JNZ        C2   NO? SET VALVE OPEN AND TEST MSB
3B6                           CO
3B7                           03
3B8                LDA        3A   YES? FETCH OTHER 1/2 OF WORD
3B9                LSTE 2     55
3BA                           02
3BB                CPI        FE   IS OTHER HALF ZERO?
3BC                           00
3BD     RZC:       JZ         CA   YES? LOOP TO START
3BE                           80
3BF                           02
3C0     NON ZRO:   MVIA       3E   ERROR NOT ZERO
3C1                           80   SET VALVE  BIT
3C2                OUT        D3   OPEN VALVE
3C3                           6F
3C4     MSB:       LDA        3A   FETCH LAST TABLE ENTRY
3C5                LSTE 1     54
3C6                           02
3C7                RLC        07   SHIFT MSB TO CARRY
3C8                JC         DA   MSB SET? LOOP TO DECR
3C9                DECR       23
3CA                           03
3CB                JMP        C3   MSB NOT SET, LOOP TO INCR
3CC                INCR       08
3CD                           03
3CE                NOP        00
3CF                NOP        80

RESTART TIME OUT LOOP

3D0     TIMOUT:    MVIA       3E   LOAD A REGISTER WITH ZERO
3D1                           00
3D2                OUT        D3   OUTPUT PM 1
3D3                PM 1       69
3D4                OUT        D3   OUTPUT PM 2
3D5                PM 2       6A
3D6                MVIA       3E   INITIALIZE COUNTER
3D7                           20
3D8                STA        32   SAVE AS CTR NO. 4
3D9                CTR 4      4B
3DA                           00
3DB     AGAIN:     LDA        3A   FETCH CTR NO. 1
3DC                CTR 1      4E
3DD                           00
3DE                CPI        FE   TEST CTR NO. 1
3DF                           02   CTR NO. 1 = 2
3E0                JNZ        C2   NO? LOOP TO AGAIN
3E1                AGAIN      DB
3E2                           03
3E3                DCRA       3D   FORCE COUNTER DOWN 1 COUNT
3E4                STA        32   SAVE NEW COUNT
3E5                           4E
3E6                           00
3E7                LDA        3A   FETCH CTR NO. 4
3E8                           4B
3E9                           00
3EA                DCRA       3D   DECREMENT CTR NO. 4
3EB                STA        32   SAVE NEW COUNT
3EC                           4B
3ED                           00
3EE                CPI        FE   TEST CTR NO. 4
3EF                           00   CTR 4 = 0?
3F0                JNZ        C2   NO? LOOP TO AGAIN
3F1                AGAIN      DB
3F2                           03
3F3                JMP        C3
3F4                START      80
3F5                           02
```

Figure B-8. C-5 Iron Bird Flap Drive Program (Sheet 5)

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